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Memphis

Organized 1944

Headquarters, Memphis

President.....W. L. Wellford, Jr.
Vice-President.....C. S. Fischer
Secretary.....A. T. Bevil
Treasurer.....R. W. Johnson
Board of Governors: A. T. Bevil, C. S. Fischer, T. O. S. Humphrey, Jr., R. W. Johnson, J. B. Lammons, N. C. Ledbetter, E. E. Scott, W. L. Wellford, Jr.

Michigan

Organized 1916

Headquarters, Detroit

President.....L. A. Burch
Vice-President.....C. F. Donohoe
Secretary.....J. H. Spurgeon
Treasurer.....G. W. Akers
Board of Governors: E. H. Clark, E. F. Hyde, T. H. Mabley, R. H. Oberschulte

Minnesota

Organized 1918

Headquarters, Minneapolis

President.....R. C. Jordan
Vice-President.....G. M. Kendrick
Secretary.....E. F. Snyder, Jr.
Treasurer.....William Sturm
Board of Governors: C. L. Benson, C. P. Peterson, O. L. Lilja

Montreal

Organized 1936

Headquarters, Montreal, Que.

President.....S. W. Salter
Vice-President.....R. R. Noyes
Secretary.....J. G. Chenevert
Treasurer.....T. G. Anglin
Board of Governors: Leo Garneau, B. J. Horsburgh, R. S. Libby, F. G. Phipps, S. R. Plamondon

Nebraska

Organized 1940

Headquarters, Omaha

President.....C. A. Carter
Vice-President.....G. W. Colburn
Secretary.....K. E. Martin
Treasurer.....S. W. Black
Board of Governors: S. W. Black, C. A. Carter, G. W. Colburn, K. R. Magarrell, K. E. Martin, B. G. Peterson, O. J. Smith

New York

Organized 1911

Headquarters, New York

President.....H. S. Johnson
Vice-President.....Carl F. Kayan
Secretary.....Carl H. Flink
Treasurer.....W. M. Heebner
Board of Governors: A. A. Bearman, M. C. Giannini, P. B. Gordon, Ernst Graber, J. E. Schechter

North Carolina

Organized 1939

Headquarters, Durham

President.....DeParx Stimson
Vice-President.....M. F. DuChateau
Secretary-Treasurer.....R. B. Crosland, Jr.
Board of Governors: E. S. Dewitt, R. O. McGary, S. J. Nicholson

North Texas

Organized 1938

Headquarters, Dallas

President.....G. A. Linskie
Vice-President.....Herman Blum, Jr.
Secretary.....A. B. Ulrich, Jr.
Treasurer.....R. E. Allison
Board of Governors: J. P. Ashcraft, B. S. Foss, Jr., C. Rollins Gardner, E. T. Gessell

Northeastern Oklahoma

Organized 1948

Headquarters, Tulsa

President.....W. R. Lee
Vice-President.....A. D. Holmes
Secretary.....R. F. Shoemaker
Treasurer.....J. N. Watt
Board of Governors: Edwin Jones, L. S. Reagan, W. C. Roads

Northern Ohio

Organized 1916

Headquarters, Cleveland

President.....R. L. Byers
Vice-President.....W. M. Rowe
Secretary.....G. V. Parmelee
Treasurer.....R. E. Sherman
Board of Governors: D. E. Mannen, Jr., W. R. Moore, John Richmond

Oklahoma

Organized 1935

Headquarters, Oklahoma City

President.....H. S. Shafer
Vice-President.....J. R. Patten
Secretary-Treasurer.....R. E. Swan
Board of Governors: G. T. Donceel, W. W. Frankfurt, J. H. Spaan, Jr.

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* Filled unexpired term.

LOCAL CHAPTER OFFICERS—1949 *(continued)*

Ontario

Organized 1922

Headquarters, Toronto

President....A. J. Strain
Vice-President....J. H. Fox
Secretary-Treasurer....H. R. Roth
Board of Governors: H. A. Baxter, N. W. Kingsland, William Philip, E. G. Spall

Oregon

Organized 1939

Headquarters, Portland

President....J. P. McDermott
Vice-President....R. C. Chewning
Secretary....K. H. Hanson
Treasurer....A. C. Lark
Board of Governors: K. N. Flocke, A. N. Hoss, E. R. Lokey, W. A. Simpson

Pacific Northwest

Organized 1928

Headquarters, Seattle, Wash.

President....J. D. Sparks
Vice-President....C. A. Pangborn
Secretary....W. B. Pride
Treasurer....R. R. Kirkwood
Board of Governors: L. L. Bysom, R. R. Kirkwood, E. H. Langdon, R. D. Morse, C. A. Pangborn, W. B. Pride, J. D. Sparks

Philadelphia

Organized 1916

Headquarters, Philadelphia

President....F. H. Buzzard
1st Vice-President....J. W. McElgin
2nd Vice-President....E. K. Wagner
Secretary....C. F. Dietz
Treasurer....M. E. Barnard
Board of Governors: M. E. Barnard, F. H. Buzzard, L. M. Church, C. F. Dietz, J. O. Kirkbride, J. W. McElgin, E. K. Wagner

Pittsburgh

Organized 1919

Headquarters, Pittsburgh

President....H. J. Kirkendall
Vice-President....A. F. Metzger
Secretary....E. H. Riesmeyer, Jr.
Treasurer....B. B. Reilly
Board of Governors: H. E. Park, W. D. Simpson, B. R. Small

Rocky Mountain

Organized 1944

Headquarters, Denver, Colo.

President....E. J. McEahern
Vice-President....R. W. Peterson
Secretary....N. H. Brickham
Treasurer....A. W. Cooper
Board of Governors: F. C. Allen, B. A. Brickham, Fred Janssen

St. Louis

Organized 1918

Headquarters, St. Louis

President....W. A. Russell
1st Vice-President....J. S. Rosebrough
2nd Vice-President....C. H. Burnap
Secretary....H. C. Sharp
Treasurer....L. L. Hamig
Board of Governors: G. H. Bemarkt, E. T. Clucas, B. L. Evans, G. W. Pieksen

Shreveport

Organized 1948

Headquarters, Shreveport

President....A. H. Otto
Vice-President....B. E. Segall, Jr.
Secretary....S. W. Beatty
Treasurer....R. M. Hood
Board of Governors: S. W. Beatty, W. S. Evans, W. E. FitzGerald, R. M. Hood, A. H. Otto, P. O. Rottmann, B. E. Segall, Jr.

South Texas

Organized 1938

Headquarters, Houston

President....R. J. Salinger
Vice-President....H. W. Broadwell
Secretary....G. J. Collins
Treasurer....C. L. Fleming
Board of Governors: R. E. Armstrong, E. G. Floeter, D. M. Mills, C. C. Quin, Jr.

Southern California

Organized 1930

Headquarters, Los Angeles

President....J. L. Blake
Vice-President....L. J. Helms
Secretary....M. C. Greiner
Treasurer....J. S. Earhart
Board of Governors: R. S. Farr, A. J. Hess, Maron Kennedy, J. F. Park, W. O. Stewart

Southwest Texas

Organized 1946

Headquarters, San Antonio

President....G. R. Rhine
Vice-President....L. S. Pawkett
Secretary-Treasurer....I. W. Wilke
Board of Governors: R. W. Barnes, F. C. Benham, Jr., R. J. Bernhardt, Allan Henry, D. E. Locher, G. R. Rhine, A. J. Rummel

Utah

Organized 1944

Headquarters, Salt Lake City

President....C. E. Ferguson
Vice-President....R. H. East
Secretary-Treasurer....A. R. Curtis
Board of Governors: J. L. Ashton, E. V. Gritton, D. B. Holford, R. A. Pons, E. J. Watts

Virginia

Organized 1946

Headquarters, Norfolk

President....J. E. White
Vice-President....W. P. Robinson
Secretary....D. E. Phillips
Treasurer....W. C. Urrutia
Board of Governors: E. D. Duval, W. P. Robinson, J. J. Shanahan, R. C. Thomas

LOCAL CHAPTER OFFICERS—1949 *(continued)*

Washington, D. C.

Organized 1935

Headquarters, Washington, D. C.

President.....P. R. Achenbach
Vice-President.....F. M. Thuney
Secretary.....J. G. Muirhead
Treasurer.....R. L. Bridgman
Board of Governors: John Everetts, Jr., A. S. Gates, Jr., W. C. Jones, G. R. Walz*

Western New York

Organized 1919

Headquarters, Buffalo

President.....F. J. Weber
1st Vice-President.....J. H. Bryce
2nd Vice-President.....T. F. Killeen
Secretary.....J. M. Quackenbush
Treasurer.....B. C. Candee
Board of Governors: M. C. Beman, Joseph Davis, W. R. Heath, S. M. Quackenbush, Edwin Woolcock

Western Michigan

Organized 1931

Headquarters, Grand Rapids

President.....J. L. Alexander
Vice-President.....V. H. Hill
Secretary.....L. A. Calcaterra
Treasurer.....J. T. Anderson
Board of Governors: W. C. DeRoo, H. R. Limbacher, H. W. Wolters

Wisconsin

Organized 1922

Headquarters, Milwaukee

President.....B. M. Kluge
Vice-President.....F. J. Nunlist, Jr.
Secretary.....J. A. Lofte
Treasurer.....H. F. Brinen
Board of Governors: E. W. Gifford, W. F. Goldsmith, W. A. Ouweneel

Student Branches

College of City of New York

Organized 1949

Headquarters, New York

President.....I. Kleinman..N. Moskowitz
Vice-President.....M. Shakun..M. H. Segner
Recording Secretary..D. Dengeles..S. Oestreicher
Corresponding Secretary.....W. R. Spencer, R. A. Alpert
Treasurer.....M. H. Segner..T. C. T. Roberts

Louisiana Polytechnic Institute

Organized 1949

Headquarters, Ruston, La.

President.....J. F. Yoder, Jr...J. C. Cox
Vice-President.....J. B. Williams..G. B. Hall
Secretary.....K. Cumings..H. D. Hayes
Treasurer.....W. M. Gill..J. R. Williams

North Carolina State College

Organized 1948

Headquarters, Raleigh, N. C.

President.....J. J. Andrews
Vice-President.....J. W. Westbrook, Jr.
Secretary.....E. A. Stroupe
Treasurer.....W. L. Glasgow
Reporter.....B. H. Cathey
Board of Governors: L. L. Vaughan, R. B. Rice

Oregon State College

Organized 1949

Headquarters, Corvallis

President.....J. S Bowen, Jr...M. B. Larson
Vice-President.....B. L. Hansen..R. D. Reese
Secretary-Treasurer.....B. W. Carkin. Helen P. Lankow

Purdue University

Organized 1948

Headquarters, W. Lafayette

Chairman.....M. E. Naftzger..C. M. Sallman
Vice-Chairman.....L. M. Zoss..P. D. Bailey
Secretary.....C. M. Wade..D. M. Williams
Reporter.....J. W. Roberts..W. L. Kixmiller
Coordinator.....C. M. Sallman..R. L. Ford

Texas A. & M. College

Organized 1946

Headquarters, College Station, Tex.

President.....R. N. Hughes
Vice-President.....W. Q. Scott
Secretary.....W. J. Bowen
Treasurer.....E. Rovinsky
Board of Governors: W. E. Long, J. S. Hopper, W. W. Caudill

University of Detroit

Organized 1949

Headquarters, Detroit

President.....L. M. Blanchette
Vice-President.....A. F. Lewis
Secretary-Treasurer.....S. H. Kosinski
Reporter.....R. E. Seaton

University of Kansas

Organized 1949

Headquarters, Lawrence, Kans.

President.....V. N. Elder
Vice-President.....J. L. Williams
Secretary-Treasurer.....L. H. Noil

University of Texas

Organized 1949

Headquarters, Austin

President.....D. S. Belvin
Vice-President.....A. G. Allen
Secretary-Treasurer.....G. G. Marwill

* Filled unexpired term.

1349

No. 1349

FIFTY-FIFTH ANNUAL MEETING, 1949

CHICAGO, ILL.

THE 55th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was held in the Stevens Hotel, Chicago, Ill., January 24-27, 1949. Establishing a new attendance record for Society meetings, 2501 persons participated in the four-day technical program, including 1325 members, 848 guests, and 330 ladies.

FIRST SESSION—MONDAY, JANUARY 24, 1949, 9:30 A.M.

Pres. G. L. Tuve opened the 55th Annual Meeting at 9:30 a.m., Monday, January 24, 1949.

REPORT OF PRESIDENT

In his report, President Tuve reviewed the efforts of the officers, committee chairmen, and executive employees, in keeping the individual member of the A.S.H.V.E. informed about Society activities.

He noted the receipt of almost 1500 membership applications in the past year, with the accompanying problems besides that of communication. We now have more chapters than the officers can visit. Our headquarters office needs more room to handle the larger volume. Our reserve funds are no longer ample for our size. Among the year's accomplishments, he gave the following:

1. Payment of the mortgage on our Research Laboratory, so that our research activities actually have a home of their own.
2. Almost a hundred committee meetings held in the Cleveland Conference rooms during the year.
3. We have been able to increase our publications program in A.S.H.V.E. while many others have been curtailing theirs, because of higher costs.
4. Our chapters and our members are becoming more conscious of their professional responsibilities in their several communities, and we are learning to cooperate with other groups of engineers, with doctors and public health authorities, with architects and owners, and with the Army and the Navy.
5. Our research program has been broadened and our Technical Advisory Committees are exerting more influence, both on the Research Program and on THE GUIDE.

Respectfully submitted,

G. L. TUVE, *President*

REPORT OF COUNCIL

During the past year five meetings of the Council were held in New York, Cleveland, Bretton Woods, Memphis, and Chicago. The officers and council members spent many long hours in meetings and discussions of Society activities and this report will summarize some of the actions taken.

The organization meeting of the 1948 Council was held in New York on February 5, and President Tuve announced the personnel of the Council and Special Committees. Action was taken on the appointment of the Secretary of the Society, the Technical Secretary, and the Assistant to the President.

The selection of depositories for Society funds was made and a certified public accountant was selected as required by the By-Laws.

During the year abstracts of Council Minutes were furnished to chapter delegates, chapter officers, and others who desired to have them.

The question of the number of Society members in the various grades received special attention during the year and also a study was made of the membership qualifications required for various grades. A new student membership application was also developed and approved.

Charters were granted for Student Branches at North Carolina State University and the Purdue University Student Branch of Indiana Chapter, and the organization of the Northeastern Oklahoma Chapter at Tulsa was completed in October.

The Council recommended to the Committee on Research an extension of the work in the physiological field and an increase in cooperative projects. Approval was given to the acceptance of Federal government sponsorship and support of research work in fundamental problems involving heating, ventilating and air conditioning, and a contract with the Navy was authorized.

An extensive survey of publication policy was made by a Special Committee and its recommendations were adopted.

Meeting programs for Bretton Woods and Chicago were approved and the invitation of the Minnesota Chapter to have the Semi-Annual Meeting 1949 at Minneapolis was accepted and the dates selected were June 20 to 22nd.

An invitation from the North Texas Chapter was approved for a meeting and Exposition at Dallas, January 23 to 26, 1950, and the invitation of the Philadelphia Chapter for the Annual Meeting and Exposition in 1951 was also approved.

The Budget for 1949 was adopted and the recommendations of the Finance Committee regarding investment of Canadian funds and the liquidation of the mortgage on the Research Laboratory was approved.

New rates for advertising space in THE GUIDE are to be effective in 1950.

The suggestion of the Guide Publication Committee to eliminate the Roll of Membership from THE GUIDE 1949 was accepted and this will be published in a separate pamphlet which will include the names of the new officers and committees.

The resignation of Prof. F. W. Hutchinson as a member of the Committee on Research was accepted with regret, and Prof. E. R. Queer was appointed to fill his term. Five candidates for election to the Committee on Research were nominated and the Council selected five members and an alternate to serve on the Nominating Committee.

The Council accepted a special contribution from Walter L. Fleisher, New York, N. Y., a past president of the Society, for a Past President's award to one of the younger members of the Society who submitted the best manuscript on some subject dealing with environment.

A comprehensive study of Society administration and organization objectives was reported and will come up for future discussion.

The Public Relations work of the Society under the direction of the Public Relations Counsel was started on November 1 and a new editorial assistant for the Journal commenced his duties December 1st.

The Council took formal action required by the Constitution and By-Laws on the acceptance of resignations, submitted by Members and cancellation of memberships because of non-payment of dues.

The past 12 months have been notable for the many projects that have been started in the interest of expanding the Society's services and in extending its influence.

Respectfully submitted,

The Council

Finance Report

R. A. Sherman, chairman of the Finance Committee, reporting on the year's activities, described the committee's efforts to revise the details of the handling of funds and of reporting the financial transactions in order to make them more easily understood.

Another important step made during the year 1948 was the return to the previous annual dues rate of $25.00 for members and associates. It is probably too early to know the full effect of this on the Society, Mr. Sherman stated. He pointed out that though it would probably mean a number of resignations, a large membership does not necessarily mean a strong Society either professionally or financially.

Complete details of the Society's income and expenditures are given in the report of the Certified public accountant which follows:

ACCOUNTANT'S REPORT

FRANK G. TUSA & CO.
Certified Public Accountants
52 William St., New York, N. Y.

The American Society of Heating and Ventilating Engineers
51 Madison Ave., New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of The American Society of Heating and Ventilating Engineers—New York, N. Y., and the related funds for the fiscal year ended October 31, 1948, and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1948. Also, for the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements; the disbursements were supported by payment vouchers and the dues income from members and the interest income from investments was accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business October 31, 1948, is submitted herewith and your attention is directed to the following comments thereon:

Cash

The cash on deposit was verified by direct communication with commercial and savings banks and reconcilement of the balances reported to us with those reflected by the books of the Society. A schedule of cash is included as part of this report.

Checks representing the cash on hand for deposit were inspected by us and the petty cash counted.

Marketable Securities

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Co., where same are deposited for safe-keeping. This asset has been included in the balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

Accounts Receivable

A trial balance of the membership dues receivable taken as of the close of business October 31, 1948, was classified as to membership and aged as follows:

MEMBERSHIP	Total
Members	$1,832.50
Associates	3,845.42
Juniors	357.50
Students	72.00
Total	$6,107.42

AGING	
1948	$5,469.92
1947	588.00
1946	36.00
1945	13.50
Total	$6,107.42

The reserves for dues and sundry accounts receivable doubtful of collection as provided in our opinion are ample.

INVENTORIES

The emblems and TRANSACTIONS on hand on October 31, 1948, were counted by us. THE GUIDE paper was verified by direct communication with the printers. All inventories were priced and computed by us.

A schedule of TRANSACTIONS inventoried follows:

Volume	Year	Quantity	Price	Amount
1-47	1895-1941	3,229	$ .40	$1,291.60
48	1942	100	1.42	142.00
49	1943	240	1.39	333.60
50	1944	197	1.82	358.54
51	1945	359	1.91	685.69
52	1946	392	1.67	654.64
		4,517		$3,466.07

DEPOSIT RECEIVABLE

The deposit placed with the United Air Lines in the sum of $425.00 was verified by direct communication.

ADVANCES

We have classified under exchanges advances made for the accommodation of council members reimbursed subsequently to the date of audit.

As at October 31, 1948, the participating employees to the retirement plan were indebted in the sum of $1,433.46 for contributions advanced by the Society to the Pension Fund. This sum is to be reimbursed to the Society by periodical deductions from wages of employees covered by the plan.

DEFERRED CHARGES

One-sixth of all subscriptions paid during the current fiscal period to *H.P.A.C.* were deferred as a future expense since the subscriptions are paid on a calendar year basis and the fiscal year of the Society ended on October 31, 1948.

TAXES

The sum of $524.20 represents Federal Income Taxes withheld from salaries paid to employees during the month of October 1948. The Society may be subject to possible Social Security tax assessments pending decision from the U. S. Treasury Department.

DUE RESEARCH

The balance due to Research Laboratory as at October 31, 1948, is made up as follows:

40% of Dues Receivable from Members and Associates		$2,162.45
Less: Depreciation charge for the current fiscal period	$1,446.42	
40% of dues prepaid by members and associates	393.24	1,839.66
BALANCE		$ 322.79

ACCRUED ACCOUNTS

The salaries and commissions have been computed and accrued in accordance with Finance Committee Report of April 8, 1948. The accrued expenses represent professional fees applicable to the current fiscal period.

Deferred Income

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follow:

Membership	Amount
Members	$ 551.51
Associates	431.59
Juniors	151.00
Students	45.00
Total	$1,179.10

Mortgage Payable

The mortgage payable on the Cleveland real estate has been verified by direct communication with the Cleveland Trust Co. On December 15, 1948, this mortgage was completely satisfied.

Reserve for Transactions

We have reserved the sum of $24,000.00 to cover the publication of Transactions, Vols. 53 and 54 which are scheduled for publication in 1949. This sum includes an increase of $6,000.00 in excess of the 1947 Budget Provision to cover the estimated cost of publishing Vol. 53.

Funds

An analysis of the following Fund Accounts covering the changes that occurred therein during the fiscal year ended October 31, 1948, is attached hereto:

General Fund
Reserve Fund
F. Paul Anderson Fund
Property Fund
Mortgage Reduction Fund

In accordance with instructions received from the Finance Committee we have set up a Property Fund account under which we have classified the Land, Buildings, Furniture, Fixtures and Library.

Respectfully submitted,

Frank G. Tuba & Co.
Certified Public Accountants

BALANCE SHEET

The American Society of Heating and Ventilating Engineers
New York, N. Y., October 31, 1948

ASSETS

General Fund			
Cash			
On Deposit		$27,101.47	
On Hand for Deposit		8,938.31	
On Hand		100.00	$36,139.78
Securities (at Cost)			
Securities (Market Value $37,113.65)		34,439.50	
Add: Accumulated Interest	$3,240.90		
Accrued Interest	125.00	3,365.90	37,805.40
Accounts Receivable			
Membership Dues	6,107.42		
Less: Reserve for Doubtful	1,521.68	4,585.74	
Advertisers and Sundry Debtors	9,671.16		
Less: Reserve for Doubtful	1,000.00	8,671.16	13,256.90
Inventories			
Transactions and Paper		3,744.71	
Emblems		328.55	
Guide Paper		1,140.00	5,213.26

BALANCE SHEET (*continued*)

DEPOSIT RECEIVABLE		425.00	
EXCHANGES		343.85	
ADVANCES TO PENSION FUND FOR EMPLOYEES		1,433.46	
DEFERRED CHARGES			
Prepaid H.P.A.C. Subscriptions		2,179.73	$ 96,797.38
PROPERTY FUND			
Land and Buildings	84,920.96		
Less: Reserve for Depreciation	3,908.07	81,012.89	
Furniture and Fixtures	3,385.01		
Less: Reserve for Depreciation	1,258.73	2,126.28	
Library		300.00	83,439.17
RESERVE FUND			
Cash on Deposit		5,134.26	
Securities at Cost (Market Value $44,865.74)	40,145.00		
Add: Accumulated Interest	4,720.74	44,865.74	50,000.00
F. PAUL ANDERSON FUND			
Cash on Deposit	60.00		
Cash on Hand for Deposit	25.00	85.00	
Securities at Cost (Market Value $947.00)	1,000.00		
Add: Accrued Interest	12.50	1,012.50	1,097.50
MORTGAGE REDUCTION FUND			
Cash on Deposit			162.78
			$231,496.83

LIABILITIES AND FUNDS

LIABILITIES				
FEDERAL WITHHOLDING TAX			$524.20	
DUE RESEARCH			322.79	
ACCRUED ACCOUNTS				
Salaries and Commissions		$6,790.15		
Expenses		800.00	7,590.15	
DEFERRED INCOME				
Prepaid Membership Dues:				
Elected Members	$1,179.10			
Candidates	614.50	1,793.60		
Prepaid Initiation Fees		1,701.94		
Past President's Award		100.00	3,595.54	
REAL ESTATE MORTGAGE PAYABLE				
Cleveland Trust Co.			7,000.00	
RESERVE FOR PUBLICATIONS				
TRANSACTIONS—Volume 53		12,000.00		
TRANSACTIONS—Volume 54		12,000.00	24,000.00	
TOTAL LIABILITIES				$43,032.68
FUNDS				
Property Fund		83,439.17		
Less: Mortgage Payable		7,000.00	76,439.17	
Reserve Fund			50,000.00	
F. Paul Anderson Fund			1,097.50	
Mortgage Reduction Fund			162.78	
General Fund			60,764.70	
NET WORTH OF SOCIETY				188,464.15
				$231,496.83

Note: This Balance Sheet is subject to possible assessments for social security taxes pending decision by the Treasury Department.

BUDGET COMPARISON—CASH RECEIPTS

The American Society of Heating and Ventilating Engineers
New York, N. Y.

For the Fiscal Year ended October 31, 1948

	Actual	Budget Provision	Increases	Decreases
Dues				
Current Members				
100—Members	$ 45,655.12	$ 45,000.00	$ 655.12	
101—Associates	40,385.60	42,000.00		$ 1,614.40
102—Juniors	3,126.50	3,500.00		373.50
103—Students	136.00	500.00		364.00
	89,303.22	91,000.00	655.12	2,351.90
New Members				
104—Members	2,165.25	4,000.00		1,834.75
105—Associates	5,429.56	5,000.00	429.56	
106—Juniors	1,507.00	500.00	1,007.00	
107—Students	829.00	500.00	329.00	
	9,930.81	10,000.00	1,765.56	1,834.75
Prior Year's Dues				
6A—Members	1,934.00	2,000.00		66.00
6B—Associates	1,931.50	2,500.00		568.50
6C—Juniors	93.00	150.00		57.00
6D—Students	—0—	—0—		—0—
	3,958.50	4,650.00	—0—	691.50
Total Dues	103,192.53	105,650.00	2,420.68	4,878.15
108—Initiation Fees	5,920.00	7,000.00		1,080.00
Deferred Income				
Prepaid Membership Dues	1,179.10	1,345.35		166.25
Prepaid Dues by Candidates	614.50	351.00	263.50	
108—Prepaid Initiation Fees	1,701.94	1,100.00	601.94	
	3,495.54	2,796.35	865.44	166.25
Publications				
115—Editorial Contract	20,166.64	20,500.00		333.36
300—Guide Advertising	54,512.95	50,000.00	4,512.95	
302—Guide Copy Sales	55,950.46	60,000.00		4,049.54
117—Transactions	1,089.59	1,000.00	89.59	
118—Books, Reprints and Codes	2,427.01	800.00	1,627.01	
	134,146.65	132,300.00	6,229.55	4,382.90
Investments Income				
125—Interest—Savings Bank	67.69	200.00		132.31
126—Interest—Securities	475.05	300.00	175.05	
127—Interest—Mortgage	242.96	250.00		7.04
	785.70	750.00	175.05	139.35
Other Income				
109—Sales of Emblems	$ 1,051.62	$ 1,000.00	$ 51.62	
110—Sales of Certificates and Frames	590.32	500.00	90.32	
81—Mortgage Retirement	2,489.84	3,500.00		1,010.16
Mortgage Receivable	6,706.56		6,706.56	
82—Business Policy Committee	1,178.83	1,200.00		21.17
206—Past President's Award	100.00		100.00	
224—Refund of Pension Fund Deposit	250.00		250.00	
	12,367.17	6,200.00	7,198.50	1,031.33
Research Laboratory Income				
34—Contributions—Earmarked	62,727.00	45,000.00	17,727.00	
33—Contributions—General	9,530.00	6,000.00	3,530.00	
Naval Research Contract		19,900.00		19,900.00
82—1948 Exposition Income	13,902.62	14,000.00		97.38
Deferred 1947 Exposition Income "A"	8,598.69	8,598.69		
Deferred 1947 Earmarked Contributions	10,018.86	10,018.86		

RESEARCH LABORATORY INCOME (*continued*)				
3A—Interest	6.69	15.00		8.31
	104,783.86	103,532.55	21,257.00	20,005.69
TOTAL RECEIPTS	$364,691.45	$358,228.90	$38,146.22	$31,683.67

Note "A"—The sum of $8,598.69 representing deferred 1947 Exposition Income is not to be considered as a cash receipt since it was collected in the prior fiscal period. It has been included only as a balancing figure herein and for the purpose of conforming with the budget.

BUDGET COMPARISON—CASH DISBURSEMENTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y., October 31, 1948

	Actual	Budget Provision	Increases	Decreases
COMMITTEES AND CHAPTERS				
150—President's Fund	$ 2,922.93	$ 2,500.00	$ 422.93	$
151—Council Travel	5,151.06	6,000.00		848.04
160—Executive Committee	105.93	100.00	5.93	
161—Finance Committee	90.29	100.00		9.71
162—Membership Committee	828.80	1,000.00		171.20
164—Standards Committee	15.89	150.00		134.11
170—Admission & Advancement Committee	526.35	600.00		73.65
171—Constitution and By-Laws	692.68	1,500.00		807.32
172—Nominating Committee	1,021.37	2,000.00		978.63
173C—Chapter Relations Committee	393.38	300.00	93.38	
173A—Chapter Speakers	2,344.94	2,000.00	344.94	
173B—Chapter Delegates Travel	3,937.35	4,000.00		62.65
201—A.S.A. Membership	100.00	100.00		
109—Emblems	1,423.19	—0—	1,423.19	
204—Membership Certificates and Frames	1,330.45	500.00	830.45	
206—Medals and Awards	209.80	250.00		40.20
	21,094.41	21,100.00	3,120.82	3,126.41
MEETINGS				
163—Meetings	3,786.80	3,500.00	286.80	
327—Chapter Meeting Allowance	900.00	900.00		
	4,686.80	4,400.00	286.80	—0—
PUBLICATIONS				
200—Journal Subscriptions	13,282.68	13,500.00		217.32
202—TRANSACTIONS—Volume 52	8,750.39	9,600.00		849.61
202—TRANSACTIONS—Volume 53		12,000.00		12,000.00
203—Membership Roll	4,115.24	2,000.00	2,115.24	
118—Books, Reprints, etc.	2,082.23		2,082.23	
305—Guide Publication and Distrib.	59,546.38	66,000.00		6,453.62
	87,776.92	103,100.00	4,197.47	19,520.55
HEADQUARTERS				
210—Salaries	59,009.44	58,500.00	509.44	
211—Public Relations Expense	5,304.34	5,000.00	304.34	
212—Travel—Secretary and Staff	2,047.18	2,000.00	47.18	
213—Rent and Light	4,638.44	4,800.00		161.56
214—Telephone	1,467.23	1,200.00	267.23	
215—Telegraph	526.64	500.00	26.64	
216—Postage	3,947.58	3,000.00	947.58	
217—General Printing	1,836.33	1,000.00	836.33	
218—Office Supplies	1,444.45	2,000.00		555.55
219—Address and Address Changes	239.60	300.00		60.40
220—Professional Services—Legal and Auditing	2,594.84	3,300.00		705.16
220A—Business Policy Committee	1,178.83	1,200.00		21.17
221—Bank Charges	55.69	100.00		44.31
11—Furniture and Fixtures	915.11	750.00	165.11	
223—General Office Expenses	2,399.97	1,500.00	899.97	
224—Pension Fund	4,226.62	2,000.00	2,226.62	
	91,832.29	87,150.00	6,230.44	1,548.15
31—MORTGAGE RETIREMENT	13,125.00	12,220.98	904.02	
RESEARCH				
30—40% of Members and Associates Dues	39,208.59	40,200.00		991.41
30A—40% of Dues Prepaid in 1947 "B"	490.20	490.20		

Research (*continued*)				
82—1948 Exposition Income	13,902.62	14,000.00		97.38
27A—Deferred—1947 Exposition Income	8,598.69	8,598.69		
34—Contributions—Earmarked	62,727.00	45,000.00	17,727.00	
33—Contributions—General	9,530.00	6,000.00	3,530.00	
Naval Research Contract	—0—	19,900.00		19,900.00
34A—Deferred 1947 Earmarked Contributions	10,018.86	10,018.86		
3A—Interest	6.69	15.00		8.31
	144,482.65	144,222.75	21,257.00	20,997.10
Totals	$362,998.07	$372,193.73	$35,996.55	$45,192.21

Note "B"—The sum of $490.20 representing 40% Dues prepaid in 1947 is not to be considered as a cash disbursement of the current fiscal period. It is included herein only as a balancing figure and in order to conform with the budget.

BALANCE SHEET

The American Society of Heating and Ventilating Engineers—Research Fund
New York, N. Y., October 31, 1948

ASSETS

Research Fund			
Cash			
On Deposit			
Treasurer's Account—Bankers Trust Co.	$32,412.85		
Director's Account—Cleveland Trust Co.	7,393.77		
Travel Account—Cleveland Trust Co.	500.00		
Thrift Account—Bank for Savings	451.51	$40,759.13	
In Transit		3,029.42	
On Hand for Deposit		2,669.40	
On Hand		250.00	$46,706.95
Accounts Receivable			
40% Dues Receivable	2,162.45		
Less: 40% Prepaid Dues $ 393.24			
Building Depreciation 1,446.42	1,839.66	322.79	
U. S. Navy Department		2,334.19	
Advances to Retirement Plan for Account of Employees		976.77	3,633.95
Permanent			
Laboratory Equipment, Furniture and Fixtures	31,393.78		
Less: Reserve for Depreciation	11,117.54	20,276.24	
Tools		300.00	20,576.24
Deferred Charges			
Unexpired Insurance Premiums			415.92
Total Research Fund			71,333.06
Endowment Fund			
Cash on Deposit			
Bank for Savings			649.41
			$71,982.47

LIABILITIES AND FUNDS

Liabilities			
Accrued Accounts			
Interest	$ 39.38		
Expenses	368.73	$ 408.11	
Deferred Income			
Heat Flow Through Glass	14,738.75		
Radiant Heating	7,780.00		
Physiological Research	2,441.00		
Sound Energy Studies	574.39		
Heat Losses Due to Infiltration	188.75	25,722.89	
Exchanges		3.29	$26,134.29

FUNDS		
Property Fund	20,576.24	
Reserve Fund	4,000.00	
Endowment Fund	649.41	
General Fund	20,622.53	45,848.18
		$71,982.47

Note "A"—This Balance Sheet may be subject to possible assessments for social security taxes pending final decision by the Treasury Department.

BUDGET COMPARISON—CASH RECEIPTS AND DISBURSEMENTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—RESEARCH FUND

New York, N. Y.

For the Fiscal Year ended October 31, 1948

	Actual	Budget Provision	Increases	Decreases
CASH RECEIPTS				
40% Dues—Members and Associates	$ 39,208.59	$ 40,690.20	$	$ 1,481.61
Heating and Ventilating Exposition—1948	13,902.62	14,000.00		97.38
Earmarked Contributions—1948 (A)	62,727.00	45,000.00	17,727.00	
General Contributions	9,530.00	6,000.00	3,530.00	
Interest	6.69	15.00		8.31
Deferred 1947 Exposition Receipts	8,598.69	8,598.69		
Office of Naval Research	—0—	19,900.00		19,900.00
TOTAL CASH RECEIPTS	133,973.59	134,203.89	21,257.00	21,487.30
Deferred 1947 Earmarked Contribution	10,018.86	10,018.86	—0—	—0—
TOTALS	$143,992.45	$144,222.75	$21,257.30	$21,487.30
EARMARKED CONTRIBUTIONS—1948 (A)				
Heat Loads and Heat Transfer including Solar Radiation Allied Studies	$26,550.00	$17,500.00	$ 9,050.00	$
Physiological Research	5,882.00	6,000.00		118.00
Air Cleaning	7,550.00	6,000.00	1,550.00	
Air Distribution	1,765.00	2,000.00		235.00
Panel Heating Studies	17,780.00	10,000.00	7,780.00	
Oil Heat Institute Project	3,200.00	3,500.00		300.00
TOTALS (A)	$62,727.00	$45,000.00	$18,380.00	$653.00

Note—The budget provision of disbursements has been amended to include the supplementary budget of the Office of Naval Research as follows:

100	—Travel of Committee	$ 1,500.00
150	—Technical Staff	3,865.00
150(A)	—Technical Staff Overhead	1,935.00
151	—Clerical Staff	400.00
151(A)	—Clerical Staff Overhead	700.00
160	—Travel Staff	1,000.00
160(A)	—Consulting Staff	500.00
208	—Building Alterations	5,000.00
300	—Equipment	5,000.00
TOTAL		$19,900.00

	Actual	Budget Provision	Increases	Decreases
CASH DISBURSEMENTS				
100—Travel and Meeting Expenses, Chairman, Research Executive Committees and Technical Advisory Committees	$ 1,493.75	$ 3,150.00	$	$ 1,656.25
101—Chairman's Office Expense	153.57	200.00		46.43
	1,647.32	3,350.00		1,702.68
LABORATORY SALARIES				
150—Administrative and Technical Staff	51,078.99	54,115.00		3,036.01
150(A)—Technical Staff Overhead	—0—	1,935.00		1,935.00
151—Clerical Staff	8,882.49	9,150.00		267.51
151(A)—Clerical Staff Overhead	—0—	700.00		700.00
152—Pension Contributions	2,715.06	1,000.00	1,715.06	
153—Part Time Technical Staff	823.17	1,500.00		676.83
	63,499.71	68,400.00	1,715.06	6,615.35

Laboratory Expenses				
160—Staff Travel	3,412.42	4,500.00		1,087.58
160(A)—Consulting Staff	—0—	500.00		500.00
161—Postage	589.02	650.00		60.98
162—Telephone and Telegraph	1,236.42	1,500.00		263.58
163—Office Supplies	1,020.85	1,000.00	20.85	
164—Mimeographing and Printing	1,306.42	1,250.00	56.42	
165—Library	246.21	250.00		3.79
166—Publications	1,712.39	2,500.00		787.61
167—Staff Insurance	126.51	200.00		73.49
168—Professional Services	250.00	250.00		
169—Unallocated	801.85	1,000.00		198.15
175—Contingencies	1,500.00	1,500.00		
250—Lab. Material and Supplies	3,819.96	4,000.00		180.04
300—Equipment (Permanent)	4,826.82	11,000.00		6,173.18
	20,848.87	30,100.00	77.27	9,328.40
Lab. Operation and Maintenance				
200—Mortgage Interest	586.29	675.00		88.71
202—Heat	1,027.68	1,200.00		172.32
203-5—Light, Power and Water	1,028.33	1,000.00	28.33	
206—Janitor Wages and Supplies	946.94	1,200.00		253.06
207—Building Maintenance	2,071.91	2,000.00	71.91	
208—Building Alterations	—0—	5,000.00		5,000.00
209—Building Insurance	217.65	200.00	17.65	
210—Legal Fees	1,759.67	1,800.00		40.33
	7,638.47	13,075.00	117.89	5,554.42
Cooperative Contract				
1000—Pledged to June 30, 1948	3,850.00	3,850.00		—0—
1100—July 1 to October 31, 1948	5,612.00	6,275.00		663.00
	9,462.00	10,125.00		663.00
	$103,096.37	$125,050.00	$1,910.22	$23,863.85

Report of Secretary

The Secretary, A. V. Hutchinson, summarized the year's work as follows:

Total membership applications 1,469.

Publication of a new booklet on the *Aims and Activities of the Society.*

Over 18,000 copies of The Guide 1948 printed, and expansion of text by about 90 pages.

Dues collections were about 97 percent. Increased dues rate for 1949 was effective.

The Secretary's office has handled all matters pertaining to meetings, production of publications, the collection of accounts, extensive correspondence with officers, committees, and chapters, operation of the Chapter Speakers Bureau, establishment of the Public Relations Program and has carried on all of the routine actions required by the Constitution and By-Laws in connection with membership, finance, contracts, and other special projects.

Respectfully submitted,

A. V. Hutchinson, *Secretary*

Annual Report of Committee on Research—1948

There were several noteworthy events in the Society's research activities for 1948. At the June meeting, the Society amended the regulations governing the Committee on Research so that research accounting and payment of bills for research expenditures could be done at the Laboratory instead of at the New York office. The accounting books for research were transferred to the Laboratory on July 1.

One of the important highlights was the trip to England by the Director of Research on behalf of the Society, to review the research that is being carried on there and to establish a channel for the exchange of research reports between the two countries.

The Director's report of this trip contains considerable information of value to the Committee on Research and to the Technical Advisory Committees.

The Committee on Research wishes to acknowledge the splendid cooperation and sympathetic consideration given by Council to research matters. The Finance Committee of Council has been especially helpful on financial and budget problems. The Committee also wishes to express appreciation to the officers of the Society for valuable help and suggestions given on research matters throughout the year.

Committee Activities

The Committee on Research held four meetings during the year. The Chairman wishes to thank the members of the Committee for their excellent cooperation and for the seriousness with which they have assumed their responsibilities.

Altogether there were 21 Technical Advisory Committee meetings held during the year. A number of these Committees met several times. Many of the meetings were held in Cleveland, at the Laboratory. An advantage in holding the meetings at the Laboratory is that the members of the staff are given an opportunity to participate in the meetings and to have personal contact with the Committee members.

The Technical Advisory Committees play an important part in our research organization in making recommendations on our research program, and suggesting authors and titles for papers and rendering assistance in the preparation of material for THE GUIDE. Their support and cooperation are appreciated, especially that of the Committee Chairmen, on whom the success of the Committee's activities depend to a large extent. It is planned to rely to an even greater extent on the Technical Advisory Committees for help in the future.

The formation of two new Technical Advisory Committees was authorized by the Committee on Research, one on Odors and one on the Heat Pump.

Support by Industry

The Committee wishes to acknowledge the splendid financial support received from industry during this past year. The number of organizations throughout the industry which participated financially increased greatly and during 1948 numbered 242. A large share of the credit for stimulating industry's interest and desire to participate in our research work is due to the activities of C. A. McKeeman, assistant to the president, and to the work of the Committee on the Promotion of Research under the chairmanship of J. E. Haines.

Research Policy

The Committee has discussed, on several occasions during this past year, the importance of determining the attitude and feelings of the Society membership toward our research program and its objectives. Consequently, the Committee requested that a letter be sent to the membership asking each member to express his views on the research activities of the Society.

The Committee wishes to express appreciation for the continued interest of the membership in our research work. Many helpful suggestions have been received from members who are not on the Committee on Research or Technical Advisory Committees.

Council has urged the Committee on Research to place greater emphasis on physiological research and cooperative research, and to study the possibilities of closer cooperation with governmental agencies. The Committee has accepted the suggestion and has increased its expenditures for physiological research, planning further increases for 1949. During this past year a contract was negotiated with the Office of Naval Research involving the Laboratory's undertaking the design, development, and con-

struction of a calorimeter for the rapid measurement and recording of heat exchange between the human body and its environment.

Many proposals involving cooperative research with the Society have been submitted by universities during this past year. Some of these proposals have been accepted and the work is already under way. A number of others have been referred to the proper Technical Advisory Committees for their study and recommendations.

Publications and Papers

The publication policy of the Society as it affects research has been discussed by a special Committee on Publication Policy appointed by the President. According to the policy adopted, each Society member is entitled to one free copy of any technical publication, including bulletins and progress reports, upon request. Notice of the availability of these publications is announced to the membership through the JOURNAL.

Future Program

The results of the recent survey of the Society membership regarding our research activities will be carefully analyzed. If this survey indicates there is an important difference of opinion between the thinking of the members of the Society and the Committee on Research or the Regulations as set forth in our Constitution, some action will have to be taken to reconcile the same. This may require a change in our policy or a determined effort to sell our members on our aims and objectives. It may prove the importance of our finding a way of informing our members more fully regarding what is going on.

On October 4, a budget of $191,550.00 for the ensuing year was submitted to Council and approved. This represents a substantial increase over last year's budget and requires increased support on the part of our members and of industry. Our enlarged research program places a greater responsibility on the members of the Committee on Research and the Technical Advisory Committees. Of the amount given, $48,700.00 is for research required by our Navy contract, and $142,850.00 is for other research.

Ways and means are being considered to make it possible for more of our Society members to participate in the planning of our research work. As an experiment at our Annual Meeting, members of the Society are being given an opportunity to attend the meetings of the Technical Advisory Committees. It is realized that this has both advantages and disadvantages, and if it should prove undesirable to continue this as a Society policy, it may be necessary to expand the membership of a number of the Technical Advisory Committees.

Laboratory Staff

The Committee on Research wishes to acknowledge the excellent cooperation of the Laboratory staff, under the able direction of Cyril Tasker, director of research. The interest shown by each staff member in his work, and the teamwork that has been demonstrated by the staff as a whole, have been very gratifying. The Committee recommended the reappointment of Mr. Tasker as Director of Research for the ensuing year, which action has been approved by Council.

An agreement has been worked out with the faculty of Case Institute of Technology whereby members of the staff of the Research Laboratory can undertake graduate studies at Case Institute. Some of the members of the staff are taking advantage of this opportunity.

The participation of the many individuals and firms who contributed* to the program of general and ear-marked projects is gratefully acknowledged.

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*See list published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March 1949, p. 121.

The Report of the Director of Research, which is appended, includes summaries of the work at the Laboratory and cooperating institutions, and of the Technical Advisory Committees.

Respectfully submitted,

L. N. HUNTER, *Chairman*
COMMITTEE ON RESEARCH

REPORT OF THE DIRECTOR OF RESEARCH

The year 1948 has seen a significant expansion of Society research both at the Cleveland Laboratory and at a number of cooperating institutions. If one can use expenditures as an index of the activity of a research program, Society research activities in 1948 increased by about 25 percent over 1947. Actually much of this expansion took place in the second half of the year when a number of new projects were initiated and others accelerated as increased industry participation made it possible to add both staff and equipment.

Perhaps the two most important steps taken in 1948 were the signing of a contract with the Office of Naval Research to undertake work in the field of physiology and human calorimetry, and the continuous activity at the Laboratory on the study of heat distribution within and behind a concrete panel, which forms an important first step in the Society's overall program on panel heating and cooling. The studies on solar radiation transmission through glass building members continued throughout the year with additional staff engaged from June onward. The program on air cleaning constituted a full-time activity in 1948 and showed signs of real progress as the year ended. We were able to initiate an experimental program on cooled ventilation jets during the summer and maintain full-time activity from October onward. An important investigation into the burning characteristics of fuel oil was undertaken under the sponsorship of the *Oil Heat Institute*.

Cooperative research was noticeably expanded in 1948 and at the year's end eight studies were in progress at six institutions. Physiological studies dealing with shock effect, under way at the College of Medicine, University of Illinois, were supported by a grant-in-aid from the U. S. Public Health Service effective as of July 1, 1948.

INSTITUTIONS COOPERATING WITH COMMITTEE ON RESEARCH

Case Institute of Technology, Cleveland, Ohio: Air Distribution by Slots and Perforated Panels; Heat Transfer of Evaporating Refrigerants.

Columbia University, New York, N. Y.: Study, by Means of the Electric Analogger, of the Effect of Idle Tubes on the Heat Flow and Temperature Field in a Concrete Slab.

Cornell University, Ithaca, N. Y.: Effects of Size of the Sol-Air Thermometer on Measurements of Sol-Air Temperature.

Kansas State College, Manhattan, Kan.: Projection of Heated and Cooled Air Streams; Studies of the Radiant Effect in Sensory Reactions of Comfort.

University of Illinois, College of Medicine, Chicago, Ill.: Physiological Adjustments of Human Beings to Sudden Change in Environment.

Michigan State College, East Lansing, Mich.: Flow Losses in Branch Take-Off Fittings.

Several phases of an overall program of research into air distribution were under attack at Case Institute of Technology, Kansas State College, and Michigan State College. The panel heating studies at the Laboratory were augmented by studies at Kansas State College and Columbia University. At Cornell University further studies were being made of the significance of various factors in solar heat gains. The U. S Weather Bureau provided basic data to be used in a cooperative study at Tulane University of sol-air temperatures for a hot and humid climate. One phase of the study of the heat transfer coefficients of evaporating refrigerants was completed at Case Institute of Technology.

Staff and Facilities

Additions made to the engineering and technical staff during 1948 brought the full-time staff at the end of the year to 23. Of these, eleven are graduate scientific staff. Additional part-time staff were used during the year as required.

Increased office facilities have been provided, improvements made to the laboratory facilities, and permanent equipment added as required, so that the laboratory inventory now stands at $20,600 after due allowances for depreciation.

Publications

Bulletin No. 1—*Heat Transmission Through Glass* came off the press at the close of the year and already a substantial number of copies have been distributed to Society members who requested both this bulletin and the two that preceded it: Bulletin No. 2 —*The Thermal Properties of Building Materials Used in Heat Flow Calculations* and Bulletin No. 3—*Forced Convection Heat Transfer from Flat Surfaces.* Seven research papers were released from the Laboratory and cooperating institutions, two were presented at the Semi-Annual Meeting and five prepared for presentation at the 55th Annual Meeting. Titles and authors are listed elsewhere.

The Guide

In addition to assistance in the review and revision of a number of chapters in THE GUIDE by Laboratory staff members, a closer liaison has been developed between the Guide Publication Committee and the various Technical Advisory Committees. H. B. Nottage, as a member of a special sub-committee, undertook a study of existing psychrometric charts and the bases on which they were developed. This Committee has recommended that the Society undertake the development of a new psychrometric chart which would lend itself more readily to the graphical solution of air conditioning problems.

Studies at the Laboratory and at Cooperating Institutions

Only the principal highlights of the projects under way at Cleveland, and at the universities cooperating with the Society in its research program, will be covered here. Fuller details will be found in research bulletins and papers or may be secured by members on request.

Air Distribution

Ventilation Jets: As the first step in a long-term plan for practical research in room air distribution, equipment for the study of chilled air jets discharging into large spaces was constructed and assembled during the latter part of the year. Instrumentation of the test equipment is proceeding in order to enable measurement to be made of temperature, velocity and turbulence intensity in the jet region.

There is still much to be done on the problem of the measurement of low velocity air streams. The Society has been instrumental in the past in adding to the sum of

knowledge in this field; the present activities are designed to investigate the significance of some of the important elements in this problem and attempt to solve them.

An extensive literature survey, covering about 150 references, undertaken by Mr. Nottage, is nearing completion. It has already proved of considerable value in defining the experimental program for the Laboratory. It is hoped that it will be possible to make this available in published form during 1949, so that it may be of lasting and time-saving value for those undertaking investigations into air distribution problems. This type of activity is one by which the Society can very properly serve the industry at large; it is seldom undertaken by industrial organizations, and even when it is carried on, the results seldom find their way into print.

Mr. Nottage has presented some views on turbulence in air distribution at the 55th Annual Meeting.

Slots: The studies on slots described at the 54th Annual Meeting were continued at Case Institute of Technology during the summer. Investigations were conducted on the air stream performance for a slot in the side of a tapered duct, the purpose being to obtain constant discharge angles along the slot in order to facilitate the application of turning vanes. Performance curves developed from theoretical considerations were checked by test results. This work is being prepared for presentation as a research paper.

Perforated Panels: Tests were also completed at Case Institute on eight perforated panels (thick transite and thin metal).

Residual velocity measurements were conducted in the constant-velocity zone near the panel, and in the zone located a short distance from the panel in which the residual velocity varies inversely with distance. The results for the residual velocities of the panels tested were found to plot on dimensionless parameters involving percentage free area, coefficient of discharge, panel area, velocity at the perforations, and distance from the panel.

Tests were conducted with air streams projected into an open space and into an enclosed space with the distance between floor and ceiling fixed. Velocity measure-

RESEARCH PAPERS—1948

1. Solar and Total Heat Gain Through Double Flat Glass, by G. V. Parmelee and W. W. Aubele (Cleveland, Ohio) (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 407).

2. Comparison of Physiological Adjustments of Human Beings During Summer and Winter, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., (Chicago, Ill.) and M. K. Fahnestock (Urbana, Ill.) (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 307).

3. Physiologic Adjustments of Normal Subjects and Cardiac Patients to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., I. R. Callen, M.D., F. K. Hick, M.D., and M. K. Fahnestock (Chicago, Ill.) (see p. 27).

4. Overall Coefficients for Flat Glass Determined Under Natural Weather Conditions, by G. V. Parmelee and W. W. Aubele (Cleveland, Ohio) (see p. 39).

5. The Internal Performance of an Induced Draft, Packed Cooling Tower, by N. W. Snyder (Berkeley, Calif.) (see p. 61).

6. Heat Transfer Rates for Refrigerant Boiling in Horizontal Tube Evaporators, by L. G. Seigel, W. L. Bryan and M. C. Huppert (Cleveland, Ohio) (see p. 83).

7. Turbulence—A Fundamental Frontier in Air Distribution, by H. B. Nottage (Cleveland, Ohio) (see p. 193).

ment was obtained with commercial-type heated thermocouple, heated thermometer, and bridled vane anemometers.

The stream profiles for perforated panels will be correlated at some future date with profiles of other type outlets in a report dealing only with stream envelopes.

Downward Projection of Heated Air: Studies reported upon at the 54th Annual Meeting continued during 1948 using the 5.75 in. convergent nozzle. Discrepancies were found between the values for entrainment obtained by direct measurement and the values calculated from heat balances. These discrepancies are being further investigated.

Tests were also made, both with room air and with heated air, using the convergent nozzle with the tubular extension of about one orifice diameter in length. Difficulties encountered in these studies show again how unsatisfactory the present situation is regarding the measurement of low velocity air streams. These difficulties show also the need for reliable methods that will ensure reproducibility.

Space is now available for the erection of less restricted test facilities in which larger orifice diameters and throws of 40 ft downward might be investigated.

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Friction in Fittings: A study of friction losses through branch take-offs in rectangular ducts is under way at Michigan State College under Dean L. G. Miller. The Laboratory has made progress on the compilation and analysis of existing data on losses of through-flow fittings. Experimental studies, unavoidably deferred throughout this year, should be under way early in 1949.

Heat Flow and Heat Transfer

Film Coefficients: Studies reported on in Research Bulletin No. 3 were extended during 1948 to cover the combined effects of natural convection and low velocity air movement over a range of 30 to 300 fpm on heat transfer from a smooth surface.

Solar Radiation Transmission: Even though weather conditions during 1948 were not quite as favorable for the studies on solar radiation transmission as they were in 1947, they constituted a full-time activity for three staff members. Two papers were prepared and Bulletin No. 1 released in published form.

Additional studies made on flat glass, in single sheets and in double glass combinations, covered transmittance for direct and diffuse solar radiation as well as total heat gain. Overall coefficients of heat transfer for single and double glass were also determined.

The investigation of solar heat transmission through glass block which commenced in May has required a considerably more detailed study than was necessary with flat glass. Four different patterns of 8 in. block were studied during the latter half of the year; the investigation is continuing as weather permits; preliminary data will be available in published form by the summer of 1949.

To measure incoming radiation from all sources on the outdoor surface of the panels under test, a convection-compensated radiometer has been designed and constructed at the Laboratory and is now undergoing calibration. It is hoped by this instrument to establish the convective and radiative heat exchanges at the outer surface of the glass as accurately as they are now established for the heat exchanges between the indoor surface of a test panel and the calorimeter used in these studies.

Securing this information is essential to a better understanding of all the factors that enter into questions of heat gains or losses from sunlit glass areas. It plays an important part in the final objective of developing design data.

Industry interest in, and support for, these studies has been well maintained during the year and has greatly encouraged the staff engaged in this work.

Cooling Loads—Periodic Heat Flow: At Cornell University, Prof. C. O. Mackey is studying the effect of the size of the sol-air thermometer, which was developed by him partially under a cooperative research agreement with the Society, on the sol-air temperatures obtained under natural weather conditions. The U. S. Weather Bureau has compiled the basic data for the determination of sol-air temperatures for New Orleans. It is expected that these will be computed and made available by the summer of 1949 and thus provide these data for a hot and humid area.

It would appear as if we had reached the stage where careful experimental studies should be arranged to supplement the information developed by analytical treatment. Periodic heat flow is still but imperfectly understood and appreciated by many air conditioning engineers. Yet it is of real significance in the development of cooling load calculations.

Insulation: The studies mentioned in the 1947 report were completed during the year and plans are being made to present the results in a technical paper.

Heat Transfer Coefficients of Evaporating Refrigerants: Prof. L. G. Seigel and his colleagues of Case Institute presented at the 55th Annual Meeting a paper covering the results obtained to date in studies designed to investigate the effects on heat transfer of such factors as: loading; oil circulation with the Freon; ratio of heated length of tube to inside diameter; refrigerant flow; and temperature difference, tube to refrigerant.

Physiological Research

The studies being carried on at the College of Medicine, University of Illinois, under Dr. R. W. Keeton, received additional support during the latter half of 1948 through a grant-in-aid from the *National Institutes of Health,* U. S. Public Health Service. Two papers were prepared: one was presented at the 1948 Semi-Annual Meeting; the second at the 55th Annual Meeting.

The first paper gave comparisons of physiological adjustments of human beings to sudden changes in environment during summer and winter; the second discusses adjustments of young normal subjects and of cardiac patients (aged 25 to 72 years) to such changes. It is of interest to learn that cardiac patients reacted in the same manner as young normal subjects to sudden changes in environmental temperature, and that no ill effects are apparent as a result of entering and leaving air conditioned spaces during the hot summer months. The studies are continuing with both healthy and physically impaired subjects of both sexes, using both special research clothing and normal summer weight clothing.

Human Calorimetry (O.N.R. Contract)

On June 20, a contract was signed with the Office of Naval Research to undertake at the Cleveland Laboratory a basic investigation in physiology. The main purposes of this investigation were:

(*a*) To develop a calorimeter for a continuous registration of human heat exchange and its components.

(*b*) To study human heat exchange and its factors under a variety of experimental conditions including extreme and sudden environmental stresses.

R. G. Huebscher of the Laboratory staff was assigned to these studies, and Dr. T. H. Benzinger, of the staff of the *Naval Medical Research Institute,* Bethesda, is

spending part of his time in Cleveland working with the Laboratory staff. The basic principle of this calorimeter was developed by Dr. Benzinger, originally at the University of Heidelberg, Germany, and later at Bethesda.

There are a great number of interesting and important engineering problems as well as problems of instrumentation involved in translating the basic principles of such a device into a full-sized calorimeter capable of holding a subject and operating to the accuracy and within the limits set up in the project outline. A special Advisory Committee was appointed under the chairmanship of Prof. G. L. Tuve, and at the end of the year sufficient progress had been made to enable the Laboratory to submit a tentative design for the full-sized calorimeter to this Committee for study and approval.

All operating costs on this project are being covered by funds made available through the Office of Naval Research.

Panel Heating and Cooling

The broad outlines of the Society's research program in panel heating were set out in the Annual Report of the Committee on Research for 1947*. There have been no serious changes in these objectives since the report was presented, and the first part of the program, under Group A, has been under way continuously ever since. The first concrete slab was poured in March, and as soon as it was dried out and the instrumentation completed, tests were begun. So far as the Laboratory staff know, the present experimental techniques, though by no means new, have not previously been applied to a problem such as this, namely the measurement of the heat flow within, behind and from a 6 ft x 6 ft concrete slab weighing over 4200 lb. Some instrumentation and control problems were therefore to be expected. They were met and overcome, and at the end of the year tests had been completed on this first slab at 4 in., 8 in., 12 in. and 16 in. tube centers, over a wide range of tube temperatures and with varying heat outputs. Tests of the effects on the heat flow of various thicknesses of insulation are currently under way. A study of floor coverings will follow. Exhaustive tests made on the first slab when operating with 4-in. tube centers gave data which made it possible to reduce the number of tests at other tube centers and under other conditions.

A second slab, much thinner, has been poured. All thermocouples have been inserted in the interior and on the surface, and the slab is now being dried out ready for testing.

Analytical studies have been carried on to parallel the experimental investigation, but so far no conclusive results have been obtained.

At Columbia University, Prof. Carl F. Kayan is investigating, by electrical analogy methods, the effect of idle tubes on the heat flow in a slab.

An extensive bibliography of articles and references on panel heating and cooling up to the end of 1947 was prepared by the Laboratory staff and made available in mimeographed form to Society members and the industry in general. A large number of copies have already been distributed. It is planned to bring this bibliography up to date yearly.

The program under way at Kansas State College consists of an analytical study of the psychrometric room to determine the radiation interchange between an object within a bare room and its environment, and a test program within the room to study the various means of measuring surface temperatures and the radiating properties of a room. Professor Tripp is now revising a report covering *Radiation Shape Factors Between a Plane Wall and a Globe Thermometer,* which he submitted for review during the year. Tests with limited objectives are under way in the psychrometric room.

*See A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 47.

During the year a Danish engineer, Vagn Korsgaard, served for a few months on the staff of the Laboratory. During that time he prepared, with the assistance of other members of the Laboratory staff, three papers entitled:

(*a*) The Necessity of Using a Directional Mean Radiant Temperature to Describe the Thermal Conditions in Rooms (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, June 1949, p. 117).

(*b*) A New Radiometer Measuring Directional Mean Radiant Temperatures (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, July 1949, p. 129).

(*c*) Resultant Temperature and Its Measurement—A Summary of Two Danish Papers (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, September 1949, p. 121).

Air Cleaning

Steady progress has been made during 1948 in the program designed to develop sound and acceptable codes for the testing and rating of air cleaning devices. Changes were made to the test equipment described and illustrated in the 1947 Annual Report, and techniques developed during the year for both velocity and dust distribution measurements were checked under a variety of operating conditions.

It would be of considerable assistance in outlining some of the further steps to be taken in this investigation and defining the degree of accuracy to be sought in filter testing if an answer could be obtained to the question, *Can the industry tell us what it wants a filter to do?* This answer should come from filter users, perhaps more than from filter manufacturers, though it is doubtful if most users know what degree of cleaning they require and can afford to demand.

Certain filter manufacturers and some large users have continued to take a great interest in and to give active support to this program during the year. Clean air is not only important to many industries but is probably of significance in questions of health. Wider participation in this program would enable a more concentrated attack to be made on the many angles involved.

Oil-Heat Institute

Work on the Combustion Reference Test Unit mentioned in the 1947 report continued during the spring of 1948. The unit was transferred to Cleveland in April and set up with improved instrumentation. Tests made with three widely different oils showed that the unit had good possibilities of evaluating the burning characteristics of fuel oils suitable for domestic use. Additional instrumentation seems desirable to insure reproducibility of test results, especially in the matter of smoke density measurement. Certain instrument makers are cooperating with the Institute and the Society to this end. Further work has been approved by the Engineering Committee of the *O.H.I.* and will be under way early in 1949.

General

The writer left the United States in mid-August and returned in mid-October after visiting a number of research stations, laboratories, colleges, and other institutions in England where studies of interest to the Society were in progress. He discussed the subject of research for the heating, ventilating and air conditioning engineer before a special meeting of the *Institution of Heating and Ventilating Engineers* on September 29 in London dealing in some detail with the Society's research program. He also spoke before the Fuel Luncheon Club on October 5 and the Rumford Club on September 23, both in London.

Liaison was established with most, if not all, of the principal investigators in the Society's field of interest, and mutual exchange of reports and memoranda arranged. This has already begun and reports, received well in advance of general publication, are proving of interest in the Society's research program.

A report, *British Research in Heating and Ventilating in 1948,* was made available in mimeographed form early in 1949. Copies may be obtained from the Laboratory.

Staff members have addressed five chapters during 1948 on various phases of the research program. Contact has been established and/or maintained with the Department of the Army and with the Navy Department.

ACKNOWLEDGMENTS

The expansion of Society research during 1948 was only possible because the Laboratory staff received every encouragement and support from the Society officers and Council and especially from the chairman and members of the Committee on Research and the members of the various Technical Advisory Committees. On behalf of the staff the writer expresses deep appreciation of this confidence. Research accounting was transferred to Cleveland as of July 1, thus enabling a closer check to be kept on project expenditures and general operating expenses. The writer gladly takes this opportunity to express his thanks to all Laboratory staff members for their loyal services and especially to Mr. Humphreys who acted as Director during his absence from the country.

Respectfully submitted,

CYRIL TASKER, Director of Research

President Tuve introduced Nathaniel Glickman, Chicago, Ill., who presented the first technical paper, Physiologic Adjustments of Normal Subjects and Cardiac Patients to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., I. R. Callen, M.D., and F. K. Hick, M.D., Chicago, Ill., and M. K. Fahnestock, Urbana, Ill. (see Chapter 1350).

G. V. Parmelee, Cleveland, Ohio, presented the second and final paper at the session, Overall Coefficients for Flat Glass Determined Under Natural Weather Conditions, by G. V. Parmelee and W. W. Aubele, Cleveland, Ohio (see Chapter 1351).

SECOND SESSION—TUESDAY, JANUARY 25, 1949, 9:45 A.M.

The Tuesday session convened at 9:45 a.m., with Pres. G. L. Tuve presiding. President Tuve called for the report of the Board of Tellers, which was presented by Roswell Farnham, chairman of the Board of Tellers, as follows:

REPORT OF TELLERS OF ELECTION

BALLOT FOR OFFICERS

	Total
President, A. E. Stacey, Jr., Syracuse, N. Y.	1453
1st Vice Pres., Lester T. Avery, Cleveland, Ohio	1454
2nd Vice Pres., L. E. Seeley, Durham, N. H.	1452
Treasurer, Ernest Szekely, Milwaukee, Wis.	1454

MEMBERS OF COUNCIL

S. W. Boyd, Atlanta, Ga.	1366
C. R. Gardner, Dallas, Texas	1373
J. D. Kroeker, Portland, Ore.	1375
C. O. Mackey, Ithaca, N. Y.	1370
Total Ballots Received	1572
Total Legal Ballots	1470
Invalid Ballots	102

COMMITTEE ON RESEARCH—THREE-YEAR TERM

	Total
I. W. Cotton, Indianapolis, Ind.	1372
Nathaniel Glickman, Chicago, Ill.	1353
W. A. Grant, Syracuse, N. Y.	1367
L. N. Hunter, Johnstown, Pa.	1368
R. D. Madison, Buffalo, N. Y.	1360

TWO-YEAR TERM

E. R. Queer, State College, Pa.	1407

Scattering votes: President....5; 2nd Vice President....1; 1st Vice President....3; Treasurer....1; Council....5; Committee on Research....6

Board of Tellers

ROSWELL FARNHAM, *Chairman*
D. B. ANDERSON
B. S. FOSS, JR.

First Vice Pres. A. E. Stacey, Jr., presided and introduced N. W. Snyder, Berkeley, Calif., author of the first paper, The Internal Performance of an Induced Draft, Packed Cooling Tower, and Mr. Snyder made the presentation (see Chapter 1352).

The second paper, Heat Transfer Rates for Refrigerant Boiling in Horizontal Tube Evaporators, by L. G. Seigel, W. L. Bryan, and M. C. Huppert, Cleveland, Ohio, was read by Mr. Seigel (see Chapter 1353).

The third and last paper at the session was Psychrometric Factors in the Air Conditioning Estimate, by C. M. Ashley, Syracuse, N. Y., and was presented by the author (see Chapter 1354).

THIRD SESSION—WEDNESDAY, JANUARY 26, 9:30 A.M.

Second Vice Pres. Lester T. Avery called the Wednesday morning session to order at 9:30 a.m. He stated that the purpose of the meeting was to bring out information about home heating, and that three presentations dealing with different phases of the subject were to be made.

W. M. Myler, Jr., Columbus, Ohio, read the paper, Are Automatic Shutters Justified on a Gas-Fired Conversion Burner, which he and H. W. Nelson, also from Columbus, prepared (see Chapter 1355).

The second paper, Performance of Fourteen Masonry Chimneys Under Steady State Conditions, by P. R. Achenbach and S. D. Cole, Washington, D. C., was given by Mr. Achenbach (see Chapter 1356).

The third and concluding paper at the session was Performance of Residential Chimneys, by L. B. Schmitt and R. B. Engdahl, Columbus, Ohio, and was presented by Mr. Schmitt (see Chapter 1363).

FOURTH SESSION—WEDNESDAY, JANUARY 26, 2:00 P.M.

At 2:00 p.m. President Tuve called the Wednesday afternoon session to order and announced that the subject would be panel heating. R. J. Lorenzi, New Haven, Conn., presented the paper, Performance of an Electrical System of Panel Heating with Four Stages of Insulation, by R. J. Lorenzi and J. F. Schreiber, Passaic, N. J. (see Chapter 1357).

C. M. Humphreys, senior engineer, A.S.H.V.E. Research Laboratory, Cleveland, Ohio, gave a short report on the progress of panel heating research conducted at the Laboratory.

The second paper, Heating a Basementless House with Radiant Baseboard, by R. H. Weigel and W. S. Harris, Urbana, Ill., was presented by Mr. Weigel (see Chapter 1358).

The third and last paper at this session was Heating Panel Time Response Study, by A. B. Algren and Ben Ciscel, Minneapolis, Minn., and was given by Mr. Ciscel (see Chapter 1359).

FIFTH SESSION—THURSDAY, JANUARY 27, 10:00 A.M.

The fifth and final technical session of the Meeting was opened by President Tuve at 10:00 a.m., and the first technical paper, Turbulence—A Fundamental Frontier in Air Distribution, by H. B. Nottage, Cleveland, Ohio (see Chapter 1360), was presented by Mr. Nottage.

The second paper, Balancing Air Delivery of a System of Manifold Air Diffusers, by G. S. Dauphinee and Peter Argentieri, New York, N. Y., was given by Mr. Argentieri (see Chapter 1361).

The third and concluding presentation was Air Distribution and Draft, by John Rydberg and Per Norbäck, Stockholm, Sweden (see Chapter 1362), and was read by C. M. Humphreys, Cleveland, Ohio.

INSTALLATION OF OFFICERS

The new officers were installed in their positions in ceremony conducted by W. H. Driscoll, past president of the Society.

Newly elected Council Members, C. Rollins Gardner, Dallas, Tex., J. Donald Kroeker, Portland, Ore., Spencer W. Boyd, Atlanta, Ga., and C. O. Mackey, Ithaca, N. Y., were introduced.

Mr. Driscoll welcomed Ernest Szekely, Milwaukee, Wis., newly elected Treasurer, and Dean Lauren E. Seeley, Durham, N. H., Second Vice President, and Lester T. Avery, Cleveland, Ohio, First Vice President.

President-elect Alfred E. Stacey was then presented by Mr. Driscoll, who said, "You have served the Society well since 1914 and I am sure that you will serve in the same fine tradition established by a long line of distinguished predecessors."

President Stacey accepted the gavel, and after expressing his deep appreciation for the honor of being elected president of the Society, said that he looked forward to the coming year with a great deal of enthusiasm.

RESOLUTIONS

Dean L. G. Miller, Lansing, Mich., chairman of the Committee on Resolutions, then presented his report as follows:

WHEREAS, Prof. G. L. Tuve has served the Society long, efficiently and faithfully on many committees, the Council, and as Vice President and President,

WHEREAS, his very able application of his talent to Society affairs, particularly research, has contributed so greatly to the advancement of the art of heating and ventilation,

BE IT RESOLVED, That the A.S.H.V.E. extends its most sincere thanks to him for his generous and unselfish efforts in behalf of this Society.

WHEREAS, this, the 55th Annual Meeting of the Society, has been one of the most outstanding Annual Meetings that the Society has ever held, not only in the all-time record attendance, but also in the quality and interest in the papers presented, in the work accomplished at the many committee meetings, in the enjoyable entertainment provided for the ladies, the members and the guests, and in the number and attraction of the exhibits at the 9th International Heating and Ventilating Exposition,

BE IT RESOLVED, That the appreciation and sincere thanks of the members of the Society be extended:

To the Illinois Chapter for the splendid manner in which they planned, organized and conducted the Annual Meeting.

To the Officers, the Council and the Staff Executives of the Society for their able direction of the Annual Meeting.

To the authors of the various technical papers presented at the Annual Meeting, for the knowledge, thought and time which they contributed in preparing and presenting their papers.

To the speakers who addressed the meeting, for their interesting and informative talks.

To those who so pleasantly entertained the members at different times during the meeting.

To the daily newspapers and the technical press for the fine reporting of the meeting.

To the management and staff of the Chicago hotels, who provided the hotel facilities for the Annual Meeting, for the splendid service and excellent cooperation.

To the manufacturers and the exhibitors at the Heating and Ventilating Exposition for the excellent displays and to Charles F. Roth and his staff for the efficient handling of the Exposition.

To the railroads and airways for assistance in transportation.

Respectfully submitted,

RESOLUTIONS COMMITTEE
LORIN G. MILLER, *Chairman*
M. C. BEMAN
G. LORNE WIGGS

On motion, properly seconded, the Resolutions were unanimously adopted by a rising vote.

The 55th Annual Meeting of the Society was adjourned at 12:00 noon on Thursday, January 27, 1949.

PROGRAM 55th ANNUAL MEETING

Stevens Hotel, Chicago, Ill.—January 24-27, 1949

Sunday—January 23

1:30 p.m. REGISTRATION—2nd Floor

Monday—January 24

8:30 a.m. REGISTRATION—(2nd Floor)

9:30 a.m. BUSINESS SESSION—(Grand Ballroom) Presiding officer: G. L. Tuve

Greetings

Reports of Officers and Committees

Report of Committee on Research, by L. N. Hunter, Chairman

Physiologic Adjustments of Normal Subjects and Cardiac Patients to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., I. R. Callen, M.D., F. K. Hick, M.D., and M. K. Fahnestock

Overall Coefficients for Flat Glass, Determined under Natural Weather Conditions, by G. V. Parmelee and W. W. Aubele

12:15 p.m. WELCOME LUNCHEON—*Speaker:* Dr. Henry T. Heald, Presiding officer: C. M. Burnam, Jr., President of Illinois Chapter
2:00 p.m. Opening 9th International Heating and Ventilating Exposition—*International Amphitheater*
4:00 p.m. Reception for Ladies-Get-Together—(2nd Floor)
9:00 p.m. Informal Party—(Grand Ballroom)

Tuesday—January 25

9:00 a.m. REGISTRATION—2nd Floor
9:30 a.m. TECHNICAL SESSION—(Grand Ballroom) Presiding officer: A. E. Stacey, Jr.
Report of Tellers of Election
The Internal Performance of an Induced Draft, Packed Cooling Tower, by N. W. Snyder
Heat Transfer Rates for Refrigerant Boiling in Horizontal Tube Evaporators, by L. G. Seigel, W. L. Bryan and M. C. Huppert
Psychrometric Factors in the Air Conditioning Estimate, by C. M. Ashley
11:30 a.m. Buses leave Stevens Hotel for Luncheon and Fashion Show at Edgewater Beach Hotel.
12:30 p.m. Luncheon Meeting—L. N. Hunter, Chairman, Committee on Research, presiding. *Speaker:* Cyril Tasker, Director of Research, A.S.H.V.E.
2:00 p.m. Inspection Trip to Merchandise Testing and Development Laboratories of Sears, Roebuck & Co.
2:30 p.m. Inspection Trip to Ilg Electric Ventilating Co. Plant
6:30 p.m. Past Presidents' Dinner—Room 4

Wednesday—January 26

9:00 a.m. REGISTRATION—2nd Floor
9:30 a.m. TECHNICAL SESSION—(Grand Ballroom) Presiding officer: Lester T. Avery
Are Automatic Air Shutters Justified on a Gas-Fired Conversion Burner?, by W. M. Myler, Jr., and H. W. Nelson
Performance of Fourteen Masonry Chimneys Under Steady State Conditions, by P. R. Achenbach and S. D. Cole
Performance of Residential Chimneys, by L. B. Schmitt and R. B. Engdahl
2:00 p.m. TECHNICAL SESSION—(Grand Ballroom) Presiding officer: G. L. Tuve
Performance of an Electrical System of Panel Heating with Four Stages of Insulation, by R. J. Lorenzi and J. F. Schreiber
Heating A Basementless House With Radiant Baseboard, by R. H. Weigel and W. S. Harris
Heating Panel Time Response Study, by A. B. Algren and Ben Ciscel
7:00 p.m. ANNUAL BANQUET
Toastmaster: John Howatt, past-president, A.S.H.V.E.
Speaker: Dr. Clark G. Kuebler, president, Ripon College, Ripon, Wis.
Subject: Formula for Freedom
Presentation of Past President's emblem by Prof. G. L. Larson

Thursday—January 27

9:00 a.m. REGISTRATION—(North Ballroom)
10:00 a.m. TECHNICAL SESSION—(North Ballroom) Presiding officer: G. L. Tuve
Turbulence—A Fundamental Frontier in Air Distribution, by H. B. Nottage

Balancing Air Delivery of a System of Manifold Air Diffusers, by G. S. Dauphinee and Peter Argentieri
Air Distribution and Draft, by John Rydberg and Per Norbäck
Unfinished Business
New Business
Resolutions
Installation of Officers
Adjournment

1:30 p.m. Inspection Trip to McDonnell & Miller Plant and Laboratory

COMMITTEE ON ARRANGEMENTS

W. A. KUECHENBERG, *General Chairman*

J. H. MILLIKEN, *Vice Chairman*

Honorary Chairmen

H. M. HART
DR. E. VERNON HILL
JOHN HOWATT
S. R. LEWIS
DR. A. C. WILLARD

Banquet Committee—H. G. Chapin, *Chairman*, M. J. Bamond, *Co-Chairman*, C. H. Bevington, C. W. DeLand, Harry Himelblau, W. G. Hoskins, C. R. Kuglin, Eugene Mathis, Orville Miller, Jr., R. A. Patterson, J. J. Philippi, S. I. Rottmayer, D. B. Scheidecker, J. F. Tobin, M. J. Stevenson.

Entertainment Committee—J. S. Kearney, *Chairman*, W. H. Brackett, A. L. Crump, M. J. Fitzgerald, E. P. Heckel, Jr., A. O. May, L. O. Paul, L. J. Potvin, C. E. Price, E. E. Sundeen, H. G. Swart.

Exposition Committee—G. W. Bornquist, *Chairman*, C. C. Graves, A. W. Lippitt, R. V. Mehaffey.

Finance Committee—P. J. Marschall, *Chairman*, W. F. Christmann, H. E. Clo.

Ladies' Committee—Mrs. W. A. Kuechenberg, *Chairman*, Mrs. C. E. Price, *Secretary*, Mrs. M. J. Bamond, Mrs. M. W. Bishop, Mrs. G. W. Bornquist, Mrs. C. M. Burnam, Jr., Mrs. H. G. Chapin, Mrs. J. S. Kearney, Mrs. P. J. Marschall, Mrs. J. H. Milliken.

Publicity Committee—O. J. Prentice, *Chairman*, W. W. Gothard, M. M. Hattis, T. V. Johnson, G. G. Turner.

Reception Committee—M. W. Bishop, *Chairman*, A. S. Ammerman, Jr., H. E. Anderson, B. L. Casey, G. B. Coffey, H. J. Couch, Thomas Cranage, C. E. Crone, J. F. Cummiskey, E. B. Ericsson, J. J. Hayes, C. W. Johnson, W. W. Lige, A. B. Martin, J. C. Matchett, J. E. McClellan, E. N. McDonnell, E. M. Mittendorff, R. O. Nelson, H. J. Prebensen, W. A. Stahl, A. G. Stout, N. W. Swanson, M. J. Van Weelden.

Sessions Committee—G. V. Zintel, *Chairman*, R. H. Burgess, J. H. Clarke, G. G. Freyder, H. G. Gragg, Herbert Kreisman, J. W. Muessig, R. W. Shields, R. M. Spencer, L. H. Streb, C. M. Vreuls, V. L. Wesby.

1350

No. 1350

PHYSIOLOGIC ADJUSTMENTS OF NORMAL SUBJECTS AND CARDIAC PATIENTS TO SUDDEN CHANGE IN ENVIRONMENT

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THE RAPIDITY and magnitude of certain physiological adjustments of healthy young adult males on exposure to sudden changes in environmental temperature and humidity have been reported[1, 2]. The adjustments occurred rapidly and placed little strain on the cardiovascular system. This is in accord with general experience of the public that no ill effects are apparent to normal individuals making the adjustments required by entering and leaving an air-conditioned space during the hot summer months. The purpose of the present study was to compare the responses of individuals having impaired cardiovascular systems with those of normal subjects when each group was exposed to sudden changes in the environmental temperature and humidity.

SUBJECTS

Ten healthy male medical students, ranging in age from 19 to 29 years (average 23 years), were the subjects for the control series. Sixteen male

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[1] Exponent numerals refer to References.

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patients with cardiovascular impairment, ranging in age from 25 to 72 years (average 57 years), were the subjects for comparison. The clinical diagnosis and functional classification of all patients, according to the *American Heart Association*[3], are presented in Table 1.

PROCEDURE, OBSERVATIONS AND EXPERIMENTAL CONDITIONS

The procedure, observations and experimental conditions were the same for the normal subjects and cardiac patients and have been described in detail in

TABLE 1. DIAGNOSIS AND CLASSIFICATION OF THE PATIENTS

DIAGNOSIS	PATIENT NUMBER	AGE	BLOOD PRESSURE	FUNCTIONAL CLASSIFICATION[a]
A. Arteriosclerotic Heart Disease				
(1) without evidence of coronary disease.	1	72	144/88	I
	2	69	174/72	II
(2) with evidence of coronary disease...	3	72	174/72	II
	4	65	166/88	II
(3) with old myocardial infarction.....	5	56	142/86	I
	6	60	140/88	II
B. Hypertensive Cardiovascular Disease				
(1) without evidence of coronary disease.	7	66	190/106	I
	8	52	232/140	I
	9	42	212/130	III
(2) with evidence of coronary disease...	10	57	162/88	II
	11	69	156/92	II
C. Hypertensive and Arteriosclerotic Heart Disease				
(1) with evidence of coronary disease...	12	62	160/88	III
D. Valvular Cardiac Disease				
(1) rheumatic etiology.....	13	45	134/94	I
	14	25	100/70	III
	15	43	148/80	I
(2) syphilitic etiology.....	16	48	138/40	III

[a] Classification according to *American Heart Association:*

"Class I. Patients...without limitation of physical activity. Ordinary physical activity causes no discomfort.

"Class II. Patient...with slight to moderate limitation of physical activity. Ordinary physical activity causes discomfort.

"Class III. Patients...with moderate to great limitation of physical activity. Less than ordinary physical activity causes discomfort.

"Class IV. Patients...unable to carry on any physical activity without discomfort."

a previous paper[1]. The subjects and patients (dressed in thermocouple union suits) remained in the comfortable room (designated comfortable room 1) for one hour and then entered and remained in the hot room for one hour, after which they returned to the comfortable room (designated comfortable room 2) for one hour.

Observations included skin and rectal temperatures, evaporative weight loss, time of onset of perspiration, blood pressure, pulse rate, and subjective sensation of warmth or coolness.

The scale for the subjective sensation of comfort was similar to that described in an earlier paper[1] and was: (1) Cold; (2) Cool; (3) Slightly cool; (4) Comfortable; (5) Slightly warm; (6) Warm; (7) Hot.

The subjects were trained in the use of the comfort scale before starting the series of experiments. Intermediate votes such as 2½, 3½, or 4½ were accepted during the adjustment period.

The comfortable room was maintained at a constant dry bulb temperature of 76 F and a water vapor pressure of 0.356 psi corresponding to a relative humidity of 80 percent and an effective temperature of 73.4. The hot room was maintained at a dry bulb temperature of 98.5 F and a water vapor pressure of

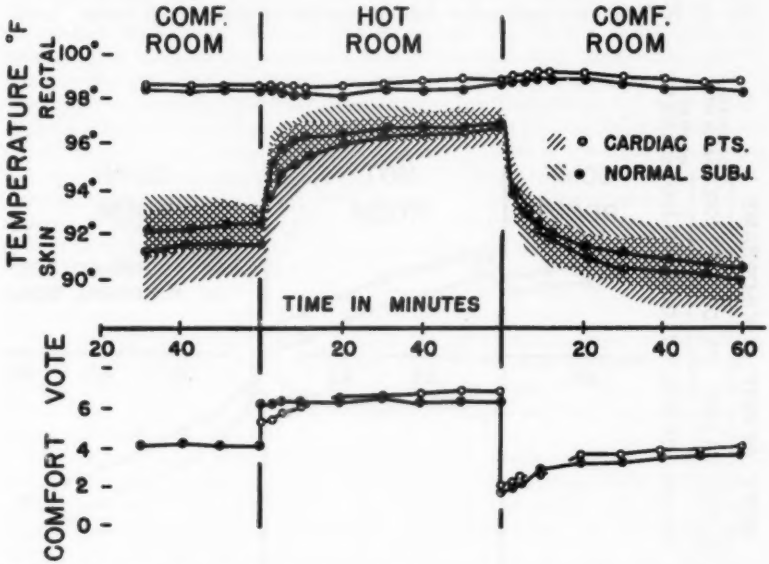

FIG. 1. AVERAGES FOR RECTAL AND MEAN SKIN TEMPERATURE AND COMFORT VOTE FOR NORMAL SUBJECTS AND CARDIAC PATIENTS IN COMFORTABLE ROOM 1, HOT ROOM AND COMFORTABLE ROOM 2

0.599 psi corresponding to a relative humidity of 66 percent and an effective temperature of 90.2.

Statistical Analysis of the Data: Fisher's modification[4] (p. 107) of Student's method for the comparison of two groups of unpaired data was used. The probabilities given were taken from Student's tables[5] and differences were considered significant when the probability (P) was less than 0.030.

RESULTS

The mean values for normal subjects and cardiac patients under comfortable conditions in equilibrium with the environment (comfortable room 1) are shown in the Figs. 1, 2, and 3. The final values for mean skin temperature averaged 92.4 F for the normal subjects and 91.5 F for the patients. The difference of 0.9 F was statistically significant ($P = 0.013$). No significant differ-

ences were found between the normal subjects and cardiac patients in final rectal temperature, final comfort vote, final pulse rate and average rate of evaporative weight loss. The blood pressure was significantly higher in the cardiac patients than in the normal subjects; the systolic pressure averaged 55.5 mm Hg higher ($P = 0.003$) and the diastolic pressure averaged 22.4 mm Hg higher ($P = 0.025$). This was anticipated because of the large difference in age.

Mean Skin Temperature: The rapidity and magnitude of the change in mean skin temperature of the normal subjects and cardiac patients can be seen in Fig. 1. Although the final mean skin temperature in comfortable room 1 aver-

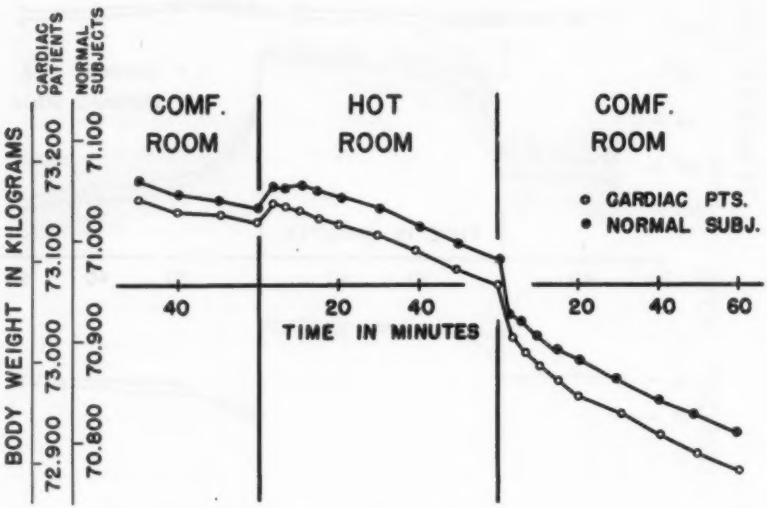

FIG. 2. AVERAGES FOR BODY WEIGHT FOR NORMAL SUBJECTS AND CARDIAC PATIENTS IN COMFORTABLE ROOM 1, HOT ROOM AND COMFORTABLE ROOM 2

aged 0.9 deg lower for the cardiac patients than for the normal subjects, the increase 10 min after entering the hot room was the same (3.8 deg) for both groups. This would suggest that the warming curve might be different. However, an analysis of the data, using only the experiments in which the final mean skin temperature in comfortable room 1 ranged from 91.4 to 93.2 F for the normal subjects (9 experiments) and 91.0 to 93.4 F for the cardiac patients (10 experiments) with averages of 92.0 F for both groups, revealed that the increase 10 min after entering the hot room was the same. At the end of the hour in the hot room there was no difference in mean skin temperature between groups. The temperature was 96.9 F for the normal subjects and 96.8 F for the cardiac patients.

In comfortable room 2, after the hot room exposure, no statistically significant differences in decrease of mean skin temperature occurred. The slight diver-

gence after 20 min (Fig. 1) merely reflects the somewhat greater, but not significant, amount of moisture present in the union suits of the cardiac patients.

The skin temperatures of various segments of the body of normal subjects and cardiac patients were compared. Only the temperature of the foot was found to be significantly different. For the normal subjects, the final comfortable room 1 temperature of the heel and toe averaged 91.6 and 88.9 F respectively, and 84.2 and 85.1 F, respectively, for the cardiac patients. Even after one hour in the hot room, the foot temperature remained definitely different. The heel and toe temperatures averaged 97.9 and 97.3 F respectively, for the normal subjects and 96.4 and 94.6 F, respectively, for the cardiac patients. At the end of the hour in comfortable room 2, however, only the toe temperature was significantly lower for the cardiac patients (90.7 as compared to 88.2 F).

Rectal Temperature: The rectal temperature has been shown [1, 2] to decrease slightly within 10 min of entering a hot environment. This is due to a transfer of heat from the deep to the superficial tissues as a result of the rapid redistribution of blood. The rectal temperature (Fig. 1) decreased within 10 min after entering the hot room, an average of 0.26 deg ($P < 0.001$) in the normal subjects and an average of 0.12 deg ($P = 0.008$) in the cardiac patients. The small difference between groups was statistically significant ($P = 0.018$) but of little clinical importance. The rectal temperature increased slightly for both groups by the end of the hour in the hot room, but there was no real difference between groups.

On entering comfortable room 2, after the hot room exposure, there was a similar small, but statistically significant, increase in rectal temperature in both groups ($P < 0.001$). This has been previously reported for normal subjects[1, 2, 6, 7]. Thereafter, the rectal temperature decreased slowly with no definite differences between groups.

Perspiration and Evaporation: The time when visible perspiration appeared in the hot room varied widely in both groups. The average time of appearance was 9.3 and 11.2 min in the normal subjects and cardiac patients, respectively. The difference was not significant and indicated that the cardiac patients could perspire as readily as the normal subjects.

The estimated water loss (perspiration[2]) during the one hour in the hot room averaged 224 gms for the normal subjects and 282 for the cadiac patients. The difference of 58 gms was not statistically significant.

There was no real difference between groups in the estimated amount of available moisture on the skin and in the union suit at the time of entrance into comfortable room 2. Further, there was no significant difference in the total amount of evaporation which occurred in the hot room and in comfortable room 2 (Fig. 2).

Pulse Rate: The final pulse rate in comfortable room 1, as shown in Fig. 3, was about the same for the normal subjects and cardiac patients. Also, the increase 3 min after entering the hot room was about the same (6.3 and 5.7 beats per minute for the normal subjects and cardiac patients, respectively). On the other hand, the normal subjects had an average increase in pulse rate of 12.0 beats per minute from the end of the comfortable room 1 to the end of the hot room exposure, whereas the cardiac patients had an average increase of only 6.1 beats per minute. The difference was definite ($P = 0.005$). Thus, the initial adjustment of the pulse rate on exposure to the hot room was the same for both groups, but the response thereafter was different. After the

initial response, the normal subjects showed a further small increase which was not shown by the cardiac patients.

On entering comfortable room 2, after the hot room exposure, the pulse rate of the normal subjects decreased an average of 10.6 beats per minute, while that of the cardiac patients decreased only 2.3 beats per minute. The difference

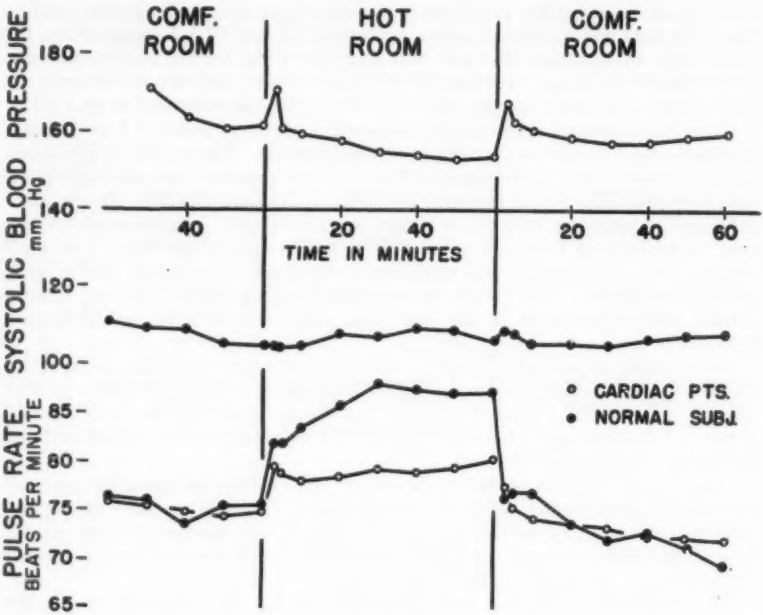

FIG. 3. AVERAGES FOR SYSTOLIC PRESSURE AND PULSE RATE FOR NORMAL SUBJECTS AND CARDIAC PATIENTS IN COMFORTABLE ROOM 1, HOT ROOM AND COMFORTABLE ROOM 2

was highly significant statistically ($P = 0.003$). Thereafter, the pulse rate in comfortable room 2 was about the same for the two groups.

Blood Pressure: The normal subjects had an average increase in systolic pressure of 3.8 mm Hg on entering the hot room and the cardiac patients an average increase of 10.5 mm Hg (Fig. 3). This difference was not significant. Further there was no real difference in the magnitude of the increase in systolic pressure on entering comfortable room 2, even though the increase averaged 16.3 mm Hg for the cardiac patients and only 7.3 mm Hg for the normal subjects.

There were no significant differences in changes of diastolic pressure at any time during the experiment.

Comfort Vote: The comfort vote immediately on entering the hot room averaged 6.2 and 5.3 for the normal subjects and cardiac patients, respectively (Fig. 1). The difference was definite ($P < 0.001$) and indicated that the subjective sensation of warmth on entering the hot room was distinctly greater for the normal subjects. However, the difference remained significant for only the first five minutes. The subjective sensation of warmth changed little for the normal subjects after entrance, whereas it continued to increase for the cardiac patients. By the end of the hour in the hot room, the cardiac patients definitely ($P = 0.023$) felt warmer than the normal subjects.

The comfort vote immediately on entering comfortable room 2 and during the remainder of the hour in this room was similar for both groups.

DISCUSSION

The final mean skin temperature in comfortable room 1 was significantly lower statistically for the cardiac patients than for the normal subjects. However, the difference amounts to only 0.9 F and clinically would be of little importance.

The lower foot temperature of the cardiac patient probably represents vascular changes incident to their disease and their age. Their blood vessels do not dilate as readily as do those of normal subjects.

There appeared to be no greater stress on the cardiovascular system of the cardiac patients than that of the normal subjects as measured by changes in pulse rate. The cardiac patients had less of an increase in pulse rate on entering the hot room, less of an overall increase during the hot room exposure and less of a decrease on entering comfortable room 2.

SUMMARY AND CONCLUSIONS

Physiological adjustments on exposure to sudden changes in environmental temperature of 16 cardiac patients were compared with those of 10 normal subjects. The data obtained justify the following conclusions:

1. The cardiac patients had a significantly lower final mean skin temperature (0.9 F) in comfortable room 1, but thereafter the changes were similar to the normal subjects.
2. The cardiac patients had a significantly lower foot temperature throughout the experiment. Other segments of the body revealed no differences.
3. With one minor exception, no differences in rectal temperature were observed.
4. No differences were observed in the average rate of evaporative weight loss. There were no differences in the time of onset of visible perspiration and in the amount of perspiration.
5. The pulse rate of the cardiac patients increased less during the hot room exposure and decreased less on entering comfortable room 2. Other adjustments were similar.
6. The changes in blood pressure were similar, even though the cardiac patients had a higher average blood pressure.
7. The subjective sensation of warmth for the cardiac patients was less on entering the hot room but became greater by the end of the hour. The comfort vote otherwise was similar.

The results indicated that the observed responses of cardiac patients on exposure to sudden changes in environmental temperature were similar in most

respects to those of young normal subjects. The few definite differences that did appear may represent differences which would have been observed as a result of the aging process in relatively normal individuals.

ACKNOWLEDGMENT

The authors gratefully acknowledge the valuable assistance of Dr. Charles Gianasi.

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DISCUSSION

CAPT. A. R. BEHNKE, Bethesda, Md. (WRITTEN): The writer suggests that the title of the paper make some mention of the fact that the normal subjects were young and that the cardiac patients were old, as well as of a change in the environment from a comfortable to a hot condition. Pulse-rate changes were of particular interest, but it would appear necessary to interpret these changes in connection with the alteration in blood pressure, particularly pulse pressure. Of special interest were the differences in skin and toe temperatures between the cardiac patients and the normal individuals.

R. W. KEETON, M.D., Chicago, Ill. (WRITTEN): Since the paper had already gone to press and also because the authors do not believe the title should stress the age differences, no change is indicated.

If consideration is given to the effects of age, then the subjects should be in a normal state of health for their respective ages in order to make valid comparisons. In the list of cardiac subjects, the ages of five were below 50 (25, 42, 43, 45, 48) and eleven were above 50. Hence, the cardiacs could not accurately be classified as *old subjects*.

It would be a most important observation if there were a correlation between pulse rate and pulse pressure. The product of pulse rate times the pulse pressure would give an indication of cardiac output. The authors did compare the changes in this product in normal patients with that in the cardiac patients and found no significant changes.

C. S. LEOPOLD, Philadelphia, Pa.: Years ago the writer asked a group of M.D.'s if there were any unusual clinical experiences, either respiratory or cardiac, of people who work in an icehouse or other occupations in which persons frequently went from chilled interiors to warm outdoors. They replied in the negative. Although they had conducted no research before answering this question, it was reasonable to assume that if the effect of such a change was not evident, the effect at most must be of a minor nature.

The writer does not believe that any of us has had first-hand, statistically significant experience of serious results with people going from a warm outdoors to a cool indoors, or vice versa. We all know someone who heard about someone who knew someone who was affected. To a large degree, this paper has put the scientific stamp on our daily observation. It is, therefore, a welcome contribution to the art.

It is unfortunate that, in spite of the data given in this paper, we shall continue to hear of the danger of this change in exposure. There is probably nothing we can do about it except to see that the Society's publications give these findings full recognition.

CYRIL TASKER, Cleveland, Ohio: It may be of interest to the membership to know that these studies are being conducted under the guidance of the Technical Advisory Committee on Physiological Research, the chairman of which for the past several years has been Dr. C.-E. A. Winslow, a past president of the Society. That Committee contains quite a number of eminent medical men and also has an international flavor. We have one member from England, one from Australia and one from France on the Committee as corresponding members.

The other rather interesting feature, somewhat apropos of what Mr. Leopold has said, is that this work has received support from two rather important bodies. The first is the *Association of American Railroads.* This body has contributed to and participated financially in this work because it is interested in the effect of sudden changes of environment on people traveling on trains. As Mr. Leopold said, we have all heard of someone who knew someone who was told about someone who got off a train in Florida and fell down on the pavement and blamed the railroads because they did not air-condition their train properly and did not gradually change train conditions as the train traveled from Cleveland, where it was nice and cold, to Florida, where it was nice and warm.

The other organization that has supported this work is the *United States Public Health Service,* through the *National Institutes of Health.* Some 18 months ago, we approached the *National Institutes of Health* with regard to this work. As soon as the results of Society work in the past were presented to that Institute and reviewed by their reviewing body, we received some financial support. That financial support is continuing. We have already made an application for its continuance through 1949 and 1950. This work therefore, is receiving the backing of important bodies, which are careful about the kind of projects into which they put their money. It is perhaps of interest to the membership to know that this work is appreciated outside the Society's own membership.

LESTER T. AVERY, Cleveland, Ohio: Since the Society has doubled its membership since 1942, many members may not be familiar with the large amount of physiological work that was done by the Society with its own funds and in its own Research Laboratory, and also in cooperation with other societies and medical schools.

A record of this work appears in the Society's TRANSACTIONS from 1925 to 1940. It provides much information on what limits people can stand. Society research was directed to the determination of two limits. One was the limit of minimum cold or the minimum heat or the minimum operation cost. The other was the maximum that the body could stand. Having determined from this research that the body is a wonderful machine, the time has come to conclude that it is not sinful to live with

air conditioning, and that persons can go from the unconditioned space to the conditioned, or vice versa, without ill effects, whether they are healthy, whether they are young, and whether they are cardiac cases or not.

If this subject needs further study, it is suggested that workers in the ice-cream industry be examined. These men go from temperatures of —40 deg to hot shipping platforms. Such a study would determine the facts and once and for all lay low this ghost of the hazard of shock effect.

The writer suspects that the complaint of shock arises comes from the man who does not want to install air conditioning. Such a person assumes air conditioning is bad because he does not want to invest in it.

Cooperative research work in the Society necessarily had to be reduced during the war years. The time has come to resume this activity. Data are now available on the maximum temperature a person can stand and continue to work. Codes have been prepared which set limits in the hot industries. The time has come for the Society to determine the optimum working conditions for these industries.

Several papers dealing with minimum strain have been discussed before this Society. The writer suggests that the next step forward is to determine, in cooperation with medical schools and hospitals, the optimum condition which can be safely recommended for living and for working. The profession would then be able to state that, for a given situation, a specific condition was good not only from the standpoint of comfort, but also from that of health.

M. K. FAHNESTOCK, Urbana, Ill.: My remarks are made, not as an author of the paper, but as a member of the Society and a person interested in physiological reactions in various environments and changes in environment. Mr. Leopold's remarks seem to indicate that there have been no difficulties experienced by persons going from one environment to another, that he has not seen any or experienced any himself, and that the paper under discussion more or less confirms his observations.

This paper is not to be regarded as an entire answer to all problems which may arise when various physiologically impaired persons pass from one environment to another. The results apply to certain types of patients exposed to certain environmental changes and under conditions of practically no activity. The writer suggests that Mr. Leopold emphasize the limitations of the experiment, stressing the significance of the types of subjects used, and the rather small number of subjects observed. The authors do not wish to give engineers the idea that their paper answers all the problems which may arise when both healthy and impaired individuals of all age groups go from one environment to another. Professor Glickman, no doubt, appreciates the fact that some cardiac patients may get into trouble when suddenly exposed to a change in environment. He certainly intends readers to appreciate the fact that the results apply only to the four types of cardiac patients defined in Table 1, and that cardiac patients are only a very small group of the total physiologically impaired population. The writer suspects that exposure to different environments of the order mentioned by Mr. Leopold could have a large effect on aggrevating chronic respiratory infections.

The magnitude and complexity of a problem which includes a general population composed of human beings of both sexes, all age groups, various activities, and different types and degrees of impairment, should be emphasized. The work reported in this paper is a start in an important area of the general problem of environment and people, a problem of concern to both the medical and engineering professions.

In the opinion of the writer, however, the interpretation and application of results of this character are largely the responsibility of medical people rather than of engineers. The work should be extended eventually to include radiation and pressure, in addition to dry bulb temperature and humidity, and the ranges of conditions should be extended. There are many situations in industry and in the armed

services where extremes in environmental factors must be taken into consideration. Length of exposure and acclimatization may be important, as in the case of the armed services where large numbers of persons may be transported by air from one climate to another in a few hours.

Experimental work of this type is costly and requires extensive physical and personnel facilities. Since it involves entire populations in one way or another, however, it should be continued and extended.

MR. LEOPOLD: With reference to Mr. Fahnestock's remarks, it should be noted that the range of temperatures, held in the experiments described in this paper, is greater than is encountered in normal work. My comments were directed to ordinary operations. Everyone realizes that it is possible to produce climatic changes which will seriously impair a person's functions.

H. E. ZIEL, Detroit, Mich.: The discussion of this paper brings up two questions: (1) Are persons, who run relatively high blood sugars, more adversely affected by sudden environmental changes than those who have cardiac conditions? and (2) Is there any way of determining blood-sugar level without taking a sample of the blood and making an analysis of it?

AUTHORS' CLOSURE BY (PROF. NATHANIEL GLICKMAN): I wish to thank the members for their stimulating discussions. Perhaps I can combine the answers to the statements made by all with the exception of the questions raised by Mr. Ziel.

First of all, we did not have large changes in the temperature gradient through which the subjects and the patients passed. In other words, they went from comfortable conditions, which you have in everyday life, to the hot conditions, which might be even a little bit hotter than the average warm, humid summer day. It was primarily under those conditions that we were trying to determine whether there were any differences between normal subjects and patients with an impaired cardiovascular system. In Table 1 of the paper, we give the clinical diagnoses of these patients. Some have rheumatic heart disease, while some have valvular disease. Those are mentioned opposite each patient, so that there is a diagnosis of the heart condition in the paper.

Now, we have done additional work. We did not feel quite satisfied that these observations gave a full answer to the problem. We felt there might be other differences. There might be other changes that went on in the body that we would not detect with our observations. Therefore, we took individuals, some of them the same as in this presentation, and had them sit in a room at a lower temperature, about 70 F, for about an hour and one-half, which was comparable to sitting in an air-conditioned movie. They then went into the hot environment and walked while carrying weights.

In other words, upon coming out of a movie and walking down the street carrying a bundle, what happens to the heart? We used the electrocardiograph to detect any changes in the electrical activity of the heart. Here, again, we found, as far as these patients were concerned, little which would indicate that there was any additional strain on these individuals. We have not yet reported on this.

Now, I would like to say that there are some individuals who are susceptible to these changes. They are individuals who would be classified as, let us say, normal, but who have a poor vaso motor tone. In other words, these individuals may be heat susceptible. They are a minority of the population. Nevertheless, they are present. A number of years ago we reported experimentally producing this socalled heat exhaustion, or fainting in a hot environment. The subjects were medical students and nurses. When the subjects were recumbent in bed, the blood pressures and heart rates were normal. In other words, the circulation was adequate while they were in that position. As soon as they had to stand in these hot conditions, a number of them

could not tolerate it and, within just a couple of minutes, became dizzy. They then either had to be helped to the bed, or, if they passed out too suddenly, had to be supported as quickly as possible to prevent them from falling to the floor. So there are a very small group of individuals who are heat susceptible.

Mr. Avery mentioned the much greater gradients in temperature, and I can agree with him that this should be investigated. We do not know. I mean, we cannot project from our present studies to much greater temperature gradients, or what are the effects of radiation on those individuals. We have to go into the problem much farther, and that will take time and we will need additional facilities.

Concerning Mr. Ziel's question, whether the individuals with a high blood sugar might be affected more adversely than the cardiac patients, I doubt it, but I certainly do not know. I presume you are thinking of diabetics when you ask that question. We hope to do some studies with diabetics and we hope to do some studies with various types of disease conditions and see whether individuals with different diseases respond differently to these sudden changes in temperature. After all, cardiacs are not the only disease states in our population. We have many other different diseases, and we have to see whether there are any ill effects on patients of that nature.

As to the last question, whether there are any methods of determining blood sugar levels without taking samples of blood, I wish I knew. There are not.

1351

No. 1351

OVERALL COEFFICIENTS FOR FLAT GLASS DETERMINED UNDER NATURAL WEATHER CONDITIONS

By GEORGE V. PARMELEE*, AND WARREN W. AUBELE**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

AN experimental investigation of heat flow through windows and glass block panels has been carried on for the past several years at the A.S.H.V.E. Research Laboratory under the direction of the Committee on Research and the Technical Advisory Committee on Heat Flow Through Glass†. Although the major part of this study has dealt with the transmittance of solar energy, a limited study has been made of overall coefficients of heat transfer for glass under natural weather conditions to assist in interpreting solar heat gain data.

Although the tests made covered a narrow range of conditions, it is believed that the results are important enough to be reported and discussed at this time. Data are given for tests made on single sheets of glass and on two sheets spaced ¼ in. apart. Wind velocities ranged from 3 to 9 mph and in several tests the glass was sheltered from the wind. Experimental data on the convection conductances for the indoor surface of the glass are given. The convection and radiation components of the outdoor surface conductance have been computed. Comparisons are made with recent guarded hot box tests.

APPARATUS

The apparatus used in the determination of the overall coefficients was a calibrated calorimeter which is being used in studies of solar heat gain through glass areas. Since it has been previously described[1], only brief mention of its

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† TAC on Heat Flow Through Glass: R. A. Miller, *chairman;* A. B. Algren, A. H. Baker, F. L. Bishop, Jr., J. E. Frazier, J. S. Herbert, E. H. Hobbie, L. K. Jones, C. O. Mackey, F. W. Preston, W. C. Randall, C. A. Richardson, Vic Sanders, H. B. Vincent, G. B. Watkins, F. C. Wood.

Presented at the 55th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1949.

[1] Exponent numerals refer to Bibliography.

TABLE 1. OVERALL COEFFICIENTS OF HEAT TRANSMISSION FOR A SINGLE SHEET OF DOUBLE STRENGTH (⅛ IN.) GLASS

DATE OF TEST	TIME EST	WIND		SKY[a] CONDITION	TEMPERATURES, F DEG				HEAT FLOW Btu per (hr) (sq ft)	U Btu per (hr) (sq ft) (F DEG)
		VELOCITY MPH	INCIDENT ANGLE, DEG		CALORIMETER		GLASS INDOOR SIDE	OUTDOOR AIR		
					SURFACE	AIR[c]				
1-20-48	7:15-8:00 p.m.	4.7	0	T.O.	73.8	72.1	46.2	36.2	35.1	0.93
1-20-48	8:15-9:15 p.m.	6.3	0	Clear	73.8	72.0	45.3	35.9	36.6	0.97
1-20-48	9:45-10:45 p.m.	5.7	0	T.O.	73.8	72.0	46.0	35.8	35.7	0.94
1-20, 21	11:45 p.m.-12:45 a.m.	6.1	45	T.O.	74.2	72.6	46.9	37.4	34.8	0.95
1-21-48	1:15-2:15 a.m.	5.3	Sheltered[b]	T.O.	74.5	73.0	46.7	37.6	32.8	0.89
1-21-48	3:15-4:00 a.m.	6.0	Sheltered[b]	T.O.	74.3	72.7	47.1	36.4	34.8	0.99
1-21-48	4:45-5:30 a.m.	4.7	Sheltered[b]	0.6C	74.3	72.8	47.3	35.7	34.3	0.89
1-21-48	Avg	5.7	0-45	T.O.	73.9	72.2	46.1	36.3	35.5	0.96
1-21-48	Avg	5.3	Sheltered[b]	T.O.	73.4	72.8	47.7	36.5	33.9	0.92
1-23-48	1:45-2:30 p.m.	5.0	45	0.5C	72.8	70.2	32.3	11.2	53.3	0.87
1-23-48	2:45-3:30 p.m.	4.3	0	0.6C	73.0	70.3	32.0	12.0	53.8	0.89
1-23-48	3:34-4:15 p.m.	3.3	20	0.5C	73.0	70.6	33.0	12.7	51.9	0.86
1-23-48	Avg	4.2	0-45	0.5C	72.9	70.4	32.4	11.9	53.0	0.87
1-26-48	2:00-2:45 p.m.	5.7	90	T.H.	76.9	74.4	38.6	24.2	49.8	0.95
1-26-48	3:00-3:45 p.m.	7.9	90	T.H.	76.8	74.2	39.1	23.8	48.9	0.92
1-26-48	Avg	6.8	90	T.H.	76.9	74.3	38.9	24.0	49.5	0.94

[a] Code: T.O., Thin Overcast; 0.5C, 0.5 Cloudiness, etc.; T.H., Thick Haze.

[b] Calorimeter rotated so that glass was in the lee of the wind.

[c] In a plane 18 in. from glass.

principal features will be made here. The calorimeter consists of a heavily-insulated blackened sheet-aluminum half-cylindrical shell to which is brazed a grid of tubes. Heat is supplied (or removed) by circulating a mixture of water and ethylene glycol through these tubes. Their arrangement and spacing is such that the calorimeter surface temperature at any point is within 1 deg of the average. This shell is bolted to the rear face of a window frame capable of being tilted through 90 deg and of orientation to any point of the compass.

A 2-ft wide border of plywood is fastened around the front face of the frame so that the test area occupies the central portion of an 8-ft square wall. The apparatus, in position for solar transmittance studies, has been shown previously[1]. The calorimeter was sealed to prevent air infiltration and contained a desiccant to control moisture migration.

Sheets of glass 48½ in. square were set into the front face of this frame and fastened to a wood strip or stop with screws. This stop allowed the glass to be set flush with the plywood panel, or to be recessed. The clear opening was 44½ in. square. The opening was double glazed by spacing the two sheets of glass with a continuous quarter-inch strip of plywood. The air space so formed was also provided with a desiccant and tightly sealed.

RESULTS OBTAINED

The results of the tests appear in Tables 1 and 2. They apply to smooth flat glass surfaces and not necessarily to actual windows. The overall coefficients of heat transmission (U values) have been based upon the difference between the temperatures of the calorimeter surface and the outdoor air and on the heat flow determined by computation of the radiation exchange and the convection heat transfer from the calorimeter to the glass. This procedure is discussed later on in this report.

As noted in the tables, some of the tests were run at night, so that it was possible to make a study of the effects of wind velocity by adjusting the orientation of the apparatus to maintain the desired average angle of incidence. Sufficient time was allowed between successive adjustments for the glass to reach a new equilibrium condition within the limits imposed by variable weather conditions.

To supplement the heat transfer data, temperature surveys of the glass surface were made with a 30 gage wire thermocouple probe. The junction and 1½ in. of each lead wire were fastened to a strip of felt ⅜ by 1½ in. in size and backed up by a pivotable wooden pressure shoe of the same size. Fig. 1 shows the results of two of these surveys. They indicate that wind velocity has a significant effect on local rates of heat flow. For parallel flow of air past a smooth heated surface, it is well known that the local convection coefficients decrease in the direction of flow. The combination of this effect with the natural circulation of air inside the calorimeter is believed to account for the temperature distribution illustrated. Computations of local radiant and convection heat transfer based upon temperatures as shown indicate approximately a 10 Btu per (hr) (sq ft) difference in heat flow between the upper left and lower right corners of the glass for the temperature pattern shown in the left hand illustration of Fig. 1.

COMPUTATION OF RESULTS

The Overall Coefficients of Heat Transmission: The overall coefficient of heat transmission, U, as it is used for computing heat losses or gains through buildings, is defined[2] as the time rate of heat flow in Btu per (hr) (sq ft) (Fahrenheit degree temperature difference between air on the inside and air on the outside of a wall, floor, roof or ceiling). The heat flow, q, in Btu per (hr) (sq ft) then is:

$$q = U\,(t_i - t_o) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where

t_i = indoor air temperature, Fahrenheit degrees.
t_o = outdoor air temperature, Fahrenheit degrees.

The heat flow, q, may also be determined by the following equation:

$$q = q_{ci} + q_{ri} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

where

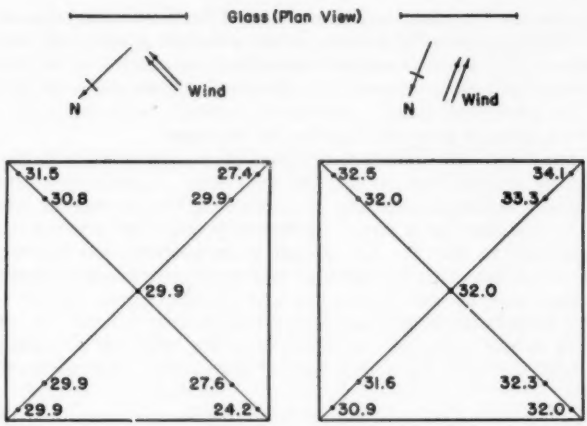
q_{ci} = the heat transferred by convection, Btu per (hour) (square foot), to the indoor surface from the air in contact with it.

q_{ri} = the net exchange by radiation, Btu per (hour) (square foot) between the indoor surface and the surrounding solid surfaces or gaseous bodies which may be capable of emitting and absorbing radiation.

TABLE 2. OVERALL COEFFICIENTS OF HEAT TRANSMISSION FOR TWO SHEETS OF ¼ IN. PLATE GLASS WITH ¼ IN. AIR SPACE

TIME[a] EST	WIND		SKY[b] CONDITION	TEMPERATURES, F DEG							HEAT FLOW BTU PER (HR) (SQ FT)	*U* BTU PER (HR) (SQ FT) (F DEG)
				CALORIMETER		INNER GLASS		OUTER GLASS		OUTDOOR AIR		
	VELOCITY MPH	INCIDENT ANGLE, DEG		SURFACE	AIR[d]	INDOOR SIDE	OUTDOOR SIDE	INDOOR SIDE	OUTDOOR SIDE			
9:45-10:30 p.m.	4.0	Sheltered[c]	O.C.	77.1	76.1	52.9	52.1	42.5	41.9	36.1	22.1	0.54
8:00-9:00 p.m.	6.8	0	O.C.	71.3	70.0	52.4	51.3	36.5	35.3	31.7	22.4	0.57
9:30-10:15 p.m.	7.7	90	O.C.	71.0	69.9	51.5	50.7	35.0	33.7	32.0	23.1	0.59
11:30 p.m.-12:30 a.m.	8.3	0	T.O.	71.5	70.8	52.3	51.7	36.5	35.2	32.6	22.6	0.58
1:15-2:15 a.m.	7.3	Sheltered[c]	0.8C	71.5	70.2	51.5	50.7	34.6	33.0	30.1	24.0	0.58
2:30-3:30 a.m.	8.2	0	O.C.	71.3	70.1	51.2	49.9	34.3	32.5	30.6	23.9	0.59
4:00-4:45 a.m.	8.8	90	O.C.	71.3	69.9	50.7	49.5	33.5	31.7	24.7	24.7	0.60
5:00-6:00 a.m.	8.8	90	0.8C	71.2	69.9	50.4	49.1	32.7	31.0	29.2	25.0	0.60
6:45-7:30 a.m.	9.0	Sheltered[c]	0.5C	71.0	69.9	49.9	48.5	32.0	30.0	27.6	25.4	0.59
Avg	8.1	0-90	O.C.	71.2	70.1	51.4	50.3	34.8	33.2	31.0	23.6	0.59
Avg	8.2	Sheltered[c]	0.7C	71.3	70.1	50.7	49.6	33.3	31.9	28.9	24.7	0.58

[a] All tests except the first made on November 25-26, 1947; first test made November 17, 1947.
[b] Code: O.C., Overcast; T.O., Thin Overcast; 0.8C, 0.8 Cloudiness, etc.
[c] Calorimeter rotated so that glass was in the lee of the wind.
[d] In a plane 18 in. from glass.



Temperature Distribution

2:15 p.m.	Time (E.S.T.)	4:15 p.m.
72.7	Calorimeter Temp, Deg F	73.1
11.1	Outside Air Temp, Deg F	12.2
29.2	Mean Glass Temp, Deg F	32.2
5.5	Wind Velocity, mph	3.4

FIG. 1. TEMPERATURE DISTRIBUTION FOR A SINGLE SHEET OF GLASS (JAN. 23, 1948)

The overall coefficients may also be defined in terms of resistances. For a single sheet of glass, the resistance equation is

$$\frac{1}{U} = R_t = \frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_i} \quad \ldots\ldots\ldots\ldots (3)$$

where

U = overall coefficient of heat transmission, Btu per (hour) (square foot) (Fahrenheit degree).

R_t = total resistance, (Fahrenheit degrees) (square foot) (hour) per Btu.

h = the surface conductance for convection and radiation combined, Btu per (hour) (square foot) (Fahrenheit degree); subscripts i and o refer to indoor and outdoor surfaces.

L = glass thickness, feet.

k = thermal conductivity of glass, Btu per (hour) (square foot) (Fahrenheit degree per foot).

To determine U experimentally, q, t_i and t_o are carefully measured and then substituted in Equation 1. In hot box tests, heat is usually supplied electrically so that q is determined from the heat equivalent of the electrical input to the guarded area. In the calorimeter apparatus q was found from separate determinations of q_{ci} and q_{ri} in Equation 2. In applying Equation 1 to the experimental

determination of U values some practical difficulties arise. In an airtight enclosure, such as a guarded hot box or the calorimeter apparatus, the average temperature of the air is dependent upon the temperatures of the surrounding solid surfaces and their respective convection conductances. If there is no artificial air movement, natural convection currents cause the air temperature to vary from point to point throughout the enclosure.

In hot box tests, heat flow is maintained by heating the air in the guarded space and by cooling the air on the cold side. Considerable stratification results, thereby leading to difficulty in determining the average air temperature, unless it is measured at a large number of points. To avoid this, fans are usually provided to stir the air enough to avoid excessive stratification. It should be noted, however, that because radiant heat exchange between the test panel surfaces and the box surfaces takes place independently of the convection effects, the temperature of the box surfaces‡ necessarily is below the ambient air temperature on the warm side and above it on the cold side by amounts dependent upon the air movement, the surface emissivities, and the surface areas involved.

In the calorimeter apparatus, the air temperatures were measured 18 in. from the test panel, as noted in Tables 1 and 2, and also by two shielded couples 4 in. from the glass. At 18 in. air temperatures were 1.0 to 2.5 deg below and at 4 in. 2.0 to 5.5 deg below the calorimeter surface temperature. Because of this variation with position and because the heat was supplied via the calorimeter surface, the overall coefficients listed in Tables 1 and 2 have been based upon the difference between the outdoor air temperature and the calorimeter surface temperature.

Computation of Heat Flow: Considerable care has been taken to define the convection and radiation conditions which govern the heat exchange between the calorimeter surface and the indoor surface of a glass sheet as installed in the apparatus. This makes it possible to compare the results with other tests and to adapt them to practical everyday surroundings. And because these conditions are known for this apparatus, heat flow through the glass has been determined by simple computations from two test observations, namely, the average temperature of the calorimeter surface and the average temperature of the indoor surface of the glass. Heat balance computations afford an independent check.

Heat flow through the glass has been determined in accordance with Equation 2. Radiation was computed from the Stefan-Boltzmann law as follows:

$$q_r = F_a F_e \, 0.173 \left[\left(\frac{T_c}{100} \right)^4 - \left(\frac{T_g}{100} \right)^4 \right] \quad \ldots\ldots\ldots \quad (4)$$

where

q_r = net heat exchange by radiation, Btu per (hour) (square foot).
F_a = the shape factor, glass with respect to calorimeter surface.
F_e = the emissivity factor, glass with respect to calorimeter surface.
T_c, T_g = the respective temperatures of the calorimeter surface and the indoor surface of the glass, degrees Rankine (Fahrenheit degrees + 460).

Because the indoor surface of the glass was completely enclosed, F_e is unity. The emissivity of the calorimeter surface has been taken as 0.97, about one

‡ The authors are unaware of data in the literature to prove this point.

percent lower than Hottel's[3] value for a flat black coating of composition similar to that used in painting the calorimeter surface. The emissivity of glass is 0.937[4]. With an area ratio of calorimeter-to-glass surface of 3.5 F_e was found from the following equation to be 0.928.

$$F_e = \frac{1}{\frac{1}{\epsilon_g} + \frac{A_g}{A_c}\left(\frac{1}{\epsilon_c} - 1\right)} \quad \ldots\ldots\ldots\ldots (5)$$

where

ϵ_c = emissivity of calorimeter surface.
ϵ_g = emissivity of glass surface.
A_c, A_g = areas of calorimeter and glass surfaces.

The data for computing heat transferred by convection were obtained from tests in which a blank panel with heat meters was substituted for the sheet of glass. Radiation exchange was computed and the convection heat transfer found by deducting this quantity from the total metered heat flow. These tests were made for the dual purpose of obtaining convection data and the unaccounted-for losses from the calorimeter and were carried out as follows. In one case, a sheet of rigid insulation board was provided with surface thermocouples and with four sets of differential thermocouples to make a heat flow meter of the blank panel. Constants for these heat meter sections were determined by calibration in a hot plate tester. A second panel was constructed of two sheets of plywood between which four commercial heat meters were sandwiched. In one instance, this was placed over the glass to allow the glass to exchange heat with the calorimeter surface and thereby duplicate the radiation conditions of actual tests. The results of these tests are shown in Fig. 2.

In order to extrapolate the convection data with more certainty, results of Griffith and Davis[5] for a 2 in. wide *enclosed* vertical air space have also been plotted on Fig. 2. This air space was about 4 ft square and therefore approximated the conditions prevailing inside the calorimeter. It has been demonstrated that convection conductances are substantially invariable for air spaces of width in excess of ¾ to 1 in. It should be noted that the calorimeter data as well as the data of Griffith and Davis are based upon the temperature difference between the two enclosing surfaces.

The heat flow values as determined by computation from experimentally observed temperatures were checked by heat balances in which the following items entered:

1. Heat supplied the calorimeter by the glycol-water mixture,
2. Metered heat leakage from the calorimeter (three sources),
3. Unaccounted-for heat leakage (determined by calibration),
4. Heat storage effects, and
5. Transmitted solar radiation (day tests only).

Values listed in Tables 1 and 2 checked values determined by heat balance within 5 percent. A number of points were discarded because of unsatisfactory checks. The daytime tests were made on a single glass sheet with orientations that avoided the direct rays of the sun. This glass transmitted 90 percent of normally incident sunlight, 82 percent of diffuse solar energy and absorbed only 2 percent of diffuse sc'ar energy. The amount of diffuse solar energy

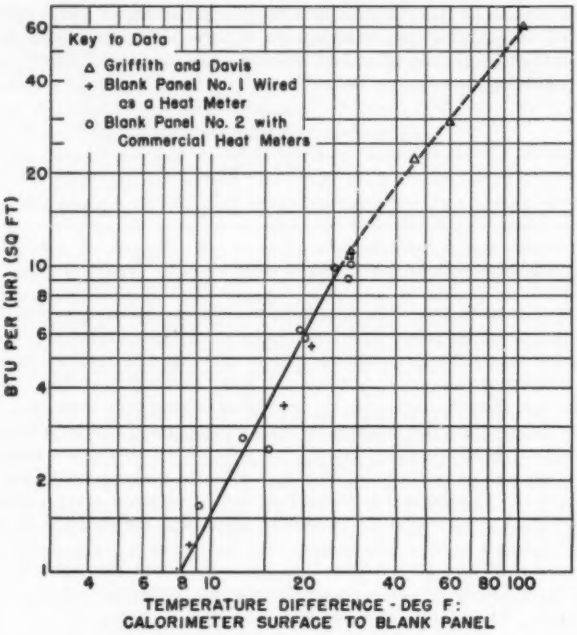


FIG. 2. CONVECTION HEAT TRANSFER AT CALORIMETER SIDE OF A BLANK PANEL; PANEL IN VERTICAL POSITION

absorbed by this glass, therefore, was so small that its effect on glass temperature, and, hence, on the computed heat flow, is considered negligible. Transmittance factors for item 5 were separately determined by calorimeter tests and other means. A sample set of data and calculations is included in Appendix A.

DISCUSSION OF RESULTS

The Significance of Surface Conductances: Windows differ from most structural elements of buildings in that the principal component, glass, is used in a thin sheet which, in itself, offers only little resistance to heat flow. The chief factors which govern the heat flow are the indoor and outdoor surface conductances and the conductance of the air space or spaces of multiple glazed windows. Also supporting members, such as sash, mullions and muntins, influence the heat flow through the glass by edge effects and disturbance of air currents, and, by virtue of their area and thermal resistance, may have an important effect on the total heat flow through the window *opening.*

The effect of the surface conductances is illustrated by Equation 3 and data taken from Table 1. For the data listed in line 8, computations show that $1/h_i$ equals 0.783 and that $1/h_o$ equals about 0.252. The first value is based upon

surface-to-surface temperature difference, the latter on surface-to-air temperature difference. The value of $\frac{L}{k}$ is approximately 0.023. In this instance the indoor resistance accounts for about 74 percent of the total thermal resistance, the outdoor surface resistance for about 24 percent, while the thermal resistance of the glass accounts for only 2 percent.

Air Space Conductance: The heat flow through the double glass installation has been checked by computing the heat flow by radiation and conduction across the ¼ in. air space. These were based on temperature measurements of the two surfaces bounding the air space. For radiation the shape factor is unity. The emissivity factor is 0.882. The conductivity of air at a mean temperature of 42 F was taken as 0.0142 Btu per (hr) (sq ft) (F deg per ft). The computed heat flow checked the total heat flow from calorimeter to glass within 5 percent. With published[6] values for combined radiation and convection across air spaces, the values checked within 3 percent.

Heat Transfer at the Outdoor Surface: The results showed that the overall coefficients for a sheltered window are slightly lower than for a window exposed to the direct wind. The differences, however, cannot be considered significant because they are within the limits of experimental error. Computation from the data presented in Tables 1 and 2 will show that the average surface conductances for the indoor surface were about 1.30 Btu per (hr) (sq ft) (F deg) for the single glass tests and 1.20 for the double glass tests. These low values mask to a considerable extent the changes in outdoor conditions. For example, a 15 percent increase in the outdoor surface conductance decreases the total thermal resistance only 4 percent. These are approximately the same changes that occur in the outdoor surface conductances and overall coefficients for the two average values of tests on single glass made on January 20-21, 1948.

It is of interest to examine heat transfer at the outdoor surface. Table 3 is an attempt to show the magnitude of the three items which comprise the total heat loss to the outdoor surroundings. The convection conductance was based

TABLE 3. SHOWING THE COMPONENTS OF THE HEAT TRANSFERRED FROM OUTDOOR GLASS TO OUTDOOR SURROUNDINGS

DATE	11-25-47	1-20-48	1-23-48	1-26-48
Temperatures, F deg				
1. Calorimeter Surface	71.2	73.9	72.9	76.9
2. Glass: Outdoor Surface	33.2	45.3[a]	31.2[a]	37.8[a]
3. Outdoor Air	31.0	36.3	12.0	24.0
4. Glass to Air	2.2	9.0	19.2	13.8
5. Outdoor Surroundings (Ground & Roof)	30[b]	35[b]	26[b]	33[b]
6. Glass to Surroundings	3.2	10.3	5.2	4.3
Atmospheric Conditions				
7. Wind Velocity, mph	8.1	5.7	4.2	6.8
8. Wind Angle of Incidence	0-90	0-45	0-45	90
9. Sky Conditions[c]	T.O.	T.O.	0.5C	T.H.
Convection Coefficient				
10. Computed (Outdoor), Btu per (hr) (sq ft) (F deg)	3.3	2.5	2.1	3.0
Heat Flow, Btu per (hr) (sq ft)				
11. Total Loss Through Glass	23.6	35.5	53.0	49.4
12. Loss by Convection to Outdoor Air	7.3	22.5	40.3	41.4
13. Loss by Radiation to Ground Surroundings	1.1	3.7	1.7	1.6
14. Items 12 + 13	8.4	26.2	42.0	43.0
15. Unaccounted-for Losses from Glass to Balance	15.2	9.3	11.0	6.4

[a] Estimated from heat flow.
[b] Estimated from observed roof temperature.
[c] Code: T.O., Thin Overcast; 0.5C, 0.5 Cloudiness; T.H., Thick Haze.

TABLE 4. RADIATION EXCHANGE BETWEEN GLASS AND ATMOSPHERE

TIME EST	TEMPERATURES, F DEG					WIND VELOCITY MPH	INCLINATION OF GLASS	INCIDENT SOLAR RADIATION BTU/ (HR) (SQ FT)	NET RADIATION EXCHANGE WITH ATMOSPHERE BTU PER (HR) (SQ FT)
	Glass Indoor Surface	Calorimeter Surface	Outdoor Air	Test House Roof					
				Sun	Shade				
10:45-11:30 a.m.	35.7	41.7	38.9	69.4	34.1	5.3	0	10	12
12:45- 1:00 p.m.	36.6	42.3	44.0	86.6	—	6.8	35	15	17
1:30- 2:00 p.m.	39.4	44.6	46.0	84.8	—	6.8	84	160	20
2:30- 3:30 p.m.	45.3	45.1	47.7	81.9	—	6.8	0	155	16
3:15- 3:45 p.m.	45.3	45.1	47.7	76.1	—	7.8	0	140	14

upon the results of a recent investigation[7] of heat transfer from a smooth surface. The surface length was conservatively taken as half the width of the glass sheet, but no allowance was made for turbulence. In all cases, the test house roof temperature was considered to be the temperature of the surroundings at ground level. The angle factors for radiant exchange between glass and ground and glass and roof are 0.32 and 0.18, respectively. It will be noted that there is a sizeable unaccounted-for loss.

Nocturnal Radiation: Natural wind is gusty and turbulent on a large scale and little or nothing is known of the quantitative effects of these characteristics on heat transfer. But the computed coefficients for convection have been based on low-turbulence wind tunnel tests of a smooth flat surface. Therefore, part of the unaccounted-for heat loss from the outdoor glass surfaces may be attributed to increased convection due to turbulence.

Part of the unaccounted-for loss, however, must be attributed to the net radiation exchange with the atmosphere, a phenomenon known as nocturnal radiation. This radiation exchange results in a cooling effect because the incoming long wave low temperature radiation from the sky, in the absence of solar effects, is insufficient to balance the outgoing radiation of a surface on the earth. In the daytime, this cooling effect usually escapes unnoticed because short wave solar radiation effects more than counterbalance the radiative deficiencies of the atmosphere.

For example, during the night tests, the roof surface and the plywood *wall* temperatures were usually a degree or two below air temperature. These temperatures invariably dropped a bit when large clear patches of sky appeared in the overcast, indicating that clearness of the sky is an important factor. More striking evidence of the significance of nocturnal radiation is demonstrated by Table 4. This table gives data that were obtained on a clear day (February 23,

1948) during the course of solar transmittance tests on a single sheet of high transmittance glass. The depression of the glass temperature below those of the outdoor air and the calorimeter surface is due to radiation loss to the sky. This loss is balanced by:

1. Convection from the outdoor air,
2. Radiation and convection from the calorimeter,
3. Solar energy absorbed (about 2 percent) by glass, and
4. Radiation from outdoor surroundings.

This table also shows the estimated net low temperature radiation exchange with the atmosphere and is believed to be conservatively estimated. Computations are shown in the Appendix B. The late afternoon data show also the effects of radiation from outdoor surroundings which, due to solar effects, are at temperatures above that of the outdoor air. For example, in the 3:15 to 3:45 p.m. period there was negligible heat exchange between calorimeter and glass, yet the glass gained heat from the outdoor air, from the test house roof, from other surroundings (the temperatures of which have been estimated) and a small amount from absorbed solar energy. The calorimeter and glass temperatures shown in Table 4, of course, are not representative of usual room temperatures, because the tests were run to obtain solar transmittance data with minimum heat leakage corrections.

Data on the magnitude of the net radiant energy exchange between ground and atmosphere are found mainly in meteorological papers. Elsasser[8] has reviewed many of these papers and gives an empirical formula proposed by Brunt for the magnitude of the incoming radiation from a clear sky to a horizontal surface. The radiation is expressed as the ratio of the incoming radiation, R_a, to black body radiation, R_s, corresponding to the air temperature near the ground.

Brunt's equation, with a correction for water vapor, is

$$\frac{R_a}{R_t} = a + b\sqrt{e} \quad \ldots\ldots\ldots\ldots\ldots \quad (6)$$

where

a, b = constants determined from observed data.
e = water vapor pressure of the air near the ground, inches of mercury.

The constants as determined by Brunt from data of a number of observers are approximately $a = 0.44$ and $b = 0.47$. Computations with these constants indicate that the net radiation loss from a horizontal surface to the atmosphere for the period 1:30 to 2:00 p.m., Table 4, should be approximately 35 Btu per (hr) (sq ft) for a clear day.

The Influence of Radiation on the Overall Coefficient: In many common cases, the usual concept of the U value will not lead to serious error in computing heat losses, provided, of course, that the surface conductances and radiant heat exchanges can be correctly calculated. Frequently, however, it happens that solid surfaces or radiating gases emit radiation at temperatures that inconveniently differ from that of the ambient air. The observations just cited in Table 4 are one example of this. In fact, the data indicate that the calorimeter surface and the outdoor air temperatures could readily be equal and yet the

calorimeter surface could, at the same time, lose heat to the glass, cooled by nocturnal radiation. Under such a circumstance, U becomes infinite.

In panel heated rooms, solid surfaces *seen* by a window commonly differ from the air temperature. Fig. 3 shows how the U value, based upon air-to-air temperature difference, may vary as the temperature of a ceiling panel varies.

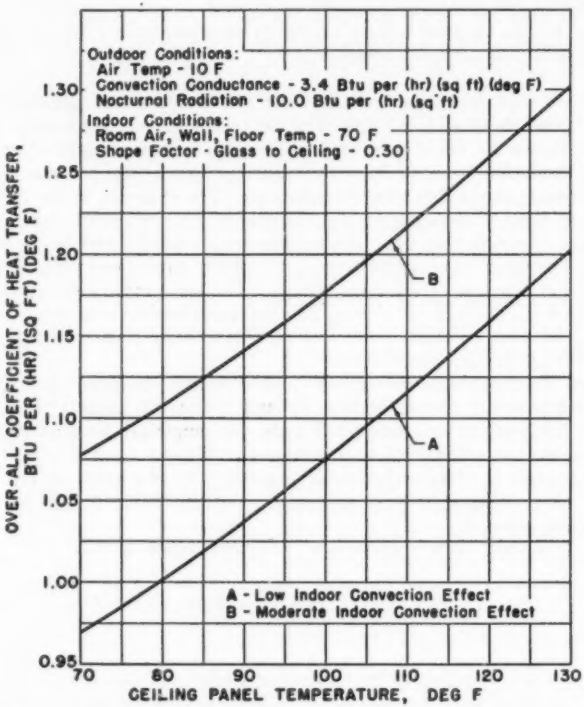


FIG. 3. CALCULATED EFFECTS OF INDOOR RADIATION AND CONVECTION CONDITIONS ON AIR-TO-AIR OVERALL COEFFICIENT FOR SINGLE GLASS

Two curves are shown. Curve A is based upon convection heat transfer assumed to vary with the temperature difference between window and indoor air, but otherwise in accordance with Fig. 2. Curve B is based upon the familiar equation[9]:

$$q_c = 0.27\ (\Delta t)^{1.25} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (7)$$

where

q_c = heat transfer by convection from air to vertical surface, Btu per (hour) (square foot).

Δt = temperature difference, Fahrenheit degrees, between air and surface.

Radiation heat exchange has been computed by Equation 4 on the assumption that the room is a black body. The authors offer the opinion that, as far as practical heated spaces are concerned, Curves A and B of Fig. 2 represent, respectively, low and moderate convection effects. A typical set of computations appears in Appendix C. All values and constants selected for these computations are arbitrary and only by chance may represent conditions chosen for design purposes. The heat flow through the glass, of course, is equal to U times (70-10).

Comparison With Other Data: Table 5 shows the results of recent[10] guarded hot box tests which included an investigation of such variables as air space width, glass thickness, light size, and air velocity. Data for 30 x 30 in. lights have been selected for comparison because this size approaches the sheet size

TABLE 5. GUARDED HOT-BOX DETERMINATIONS OF OVERALL COEFFICIENTS OF HEAT TRANSMISSION: WOOD SASH WITH 30 x 30 IN. LIGHTS

	Single Glazed Single Strength Glass		Double Glazed—¼ In. Plate Glass			
			¼ In. Air Space		½ In. Air Space	
Warm Side Air, F deg	69.7	68.0	70.8	69.7	69.0	70.8
Cold Side Air, F deg	8.2	11.1	9.8	10.3	10.3	10.3
Warm Side Surface, F deg	34.5	26.0	49.0	44.1	51.2	49.9
Cold Side Surface, F deg	29.9	21.1	26.0	17.2	22.5	14.7
Air Velocity, mph	2.5	14.4	2.5	14.8	2.8	14.3
Heat Flow, Btu per (hr) (sq ft)	54.87	67.00	31.97	38.34	27.82	35.00
U, Btu per (hr) (sq ft) F deg)	0.892	1.179	0.524	0.645	0.474	0.578

used in the calorimeter and minimizes muntin effects. The results compare favorably with guarded hot box tests on the basis of overall coefficients. By interpolation, U for a single sheet and 5.7 mph is 0.988. For two sheets of ¼ in. plate at ¼ in. air space and 8.1 mph U is 0.580. Corresponding values from the calorimeter tests of 0.95 and 0.59 are in close agreement. There was considerable difference, however, in both the radiation and convection heat transfer conditions. Nocturnal radiation, wind gustiness, and low indoor convection effects were absent from the hot box tests. On the other hand, thermal resistance and turbulence due to muntins were absent in the calorimeter tests. It might be pointed out that, after these hot box tests were completed, a honeycomb straightener was installed at the entrance to the channel through which air was passed on the cold side of the test panels. This reduced the low velocity values about 8 percent and the high velocity values about 2 percent.

The results of these tests may now be compared with commonly used[11] U values, namely 1.13 for single glass and 0.45 for double glass. The former value appears to be a reasonable value, if the radiation exchange on the outdoor side approximates black body conditions. As pointed out earlier in this paper, it may be on the low side. On the other hand, if radiation exchange between glass and indoor surroundings is at a minimum, as in sun rooms or greenhouses, the value of 1.13 may well be too high. Unusual convection con-

ditions, such as those caused by unit ventilators under windows, also need to be considered.

The value of 0.45 for double glass is clearly too low for ordinary conditions and a 15 mph wind. The value for a ¼ in. air space should be of the order of 0.60 to 0.65. For ½ in. or wider air spaces, the coefficient is shown by hot box tests to be of the order of 0.55 to 0.60.

The results of both hot box and calorimeter tests are also in good agreement with the findings of a critical review[12] of published and unpublished data on heat transmission through glass. Analysis of the results of a large number of tests on all types of windows showed that, for practical situations which reasonably approximate hot box test conditions, the presently used value for single glass is a reasonable one, but that commonly accepted U values for double and triple glazed openings were 20 to 30 percent too low.

SUMMARY

1. Tests for heat loss through single and two air-spaced sheets of flat glass show that radiation exchange between the glass on one hand and the ground surroundings and the atmosphere on the other plays a significant part in heat loss from outdoor surfaces. Because of the limited range of conditions under which these tests were made with regard to wind velocity and clearness of the atmosphere, further investigation is suggested.

2. The uncertainty connected with computing the radiation exchange and the convection loss due to turbulent wind conditions at an outdoor surface indicates the need for improved instrumentation. It is apparent that an instrument is needed which is capable of determining the magnitude of the net low temperature radiant energy exchange between a surface at the earth and the atmosphere. This instrument should be adaptable for both nocturnal and daylight conditions.

3. It has been demonstrated that the U value concept as used in estimating building heat losses or gains does not satisfactorily take into account many common radiation exchange conditions. The significant effect of indoor conditions on heat transfer to a window suggests the need for field studies of heat loss through such openings under different typical surroundings.

4. Although the conditions which governed heat loss in the calorimeter tests do not necessarily duplicate the conditions to be found in practical situations, the results indicate that for many cases the currently used U value of 1.13 for single glass is a good approximation but that the value of 0.45 for double glazed windows is from 25 to 45 percent too low for design wind velocities. This statement does *not take into account the influence of sash members.*

ACKNOWLEDGMENTS

The authors acknowledge with thanks the assistance and helpful criticisms of their colleagues of the A.S.H.V.E. Research Laboratory staff. They also wish to thank the Libbey-Owens-Ford Glass Co. and the Pittsburgh Testing Laboratory for data used in this report, and the members of the Technical Advisory Committee on Heat Flow Through Glass for useful suggestions and advice.

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APPENDIX A

DETERMINATION OF OVERALL COEFFICIENT U FROM EXPERIMENTAL DATA

Average Data for January 20, 1948; 7:15 to 8:00 p.m.:

1. Temperatures, Fahrenheit:
   - *a.* Calorimeter surface (ten points) 73.8
   - *b.* Calorimeter surface at start of period 73.8
   - *c.* Calorimeter surface at end of period 73.8
   - *d.* Calorimeter air temperature at 18 in. (4 points) 72.1
   - *e.* Calorimeter air temperature at 4 in. (2 points) 70.8
   - *f.* Glass, indoor surface (4 points) 46.2
   - *g.* Outdoor air, dry bulb 36.2
   - *h.* Glycol-water 75.2
   - *i.* Calorimeter casing 38.1

2. Heat Flow Through Glass by Computation:
   - *a.* Heat flow to glass by convection at temperature difference of $73.8 - 46.2 = 27.6$. From Fig. 3, $q_c = 10.5$ Btu per (hr) (sq ft).
   - *b.* Heat flow to glass by radiation, from Equations 4 and 5:

$$q_r = 0.928 \times 1.0 \times 0.173 \left[\left(\frac{73.8 + 460}{100}\right)^4 - \left(\frac{46.2 + 460}{100}\right)^4\right]$$

$$= 24.6 \text{ Btu per (hr) (sq ft)}.$$

   - *c.* Total heat flow: $10.5 + 24.6 = 35.1$ Btu per (hr) (sq ft).
   - *d.* $U = 35.1\ (73.8 - 36.2) = 0.93$ Btu per (hr) (sq ft) (F deg).

3. Heat Flow Through Glass by Heat Balance on Calorimeter:

a. Heat supplied by glycol-water mixture:
Specific heat at 75.2 F and 51 percent glycol: 0.795 Btu per (lb) (F deg)
Flow rate at 14.58 in. Hg (by calibration): 538 lb per hr
Temperature drop: (thermopile millivolt output ÷ 24 pairs) ×
(degrees per millivolt at 75.2 F deg) = (1.038 × 45.4)/24 = 1.96 F deg
Heat supplied per sq ft of glass = (538 x 0.795 × 1.96)/13.72 = 61.1

b. Corrections for heat leakage:

Heat flow through insulation for 4.69 millivolt drop across 2 in. thickness as indicated by 17 series-connected differential couples:

$$q = \left(\text{Area} \times \frac{k}{L} \times \Delta t\right)/13.72$$

$$= \left(47.4 \times \frac{0.28}{2} \times \frac{4.69}{17} \times 45.6\right)/13.72 = 6.1 \text{ Btu per (hr) (sq ft)}$$

Heat leakage via aluminum flange for 0.238 millivolt drop across 3-in. path length as indicated by 4 series-connected differential couples:

$$q = \left(\text{Area} \times \frac{k}{L} \times \Delta t\right)/13.72$$

$$= \left(0.181 \times \frac{117}{0.25} \times \frac{0.238}{4} \times 51.1\right)/13.72 = 18.8 \text{ Btu per (hr) (sq ft)}$$

Heat leakage via corner block on which glass is set as indicated by 0.522 millivolt drop across strip type heat meters:

q = (millivolt drop) × meter constant
= 0.522 × 4.94 = 2.6 Btu per (hr) (sq ft)

Heat storage in calorimeter shell:

q = (heat capacity × temperature rise per hr) per 13.72
= (11.6 x 0.0 × 60/45)/13.72 = 0.0 Btu per (hr) (sq ft)

Unaccounted for loss (by calibration):

For temperature difference between calorimeter surface and calorimeter casing of 35.7 F deg: 0.3 Btu per (hr) (sq ft)

c. Heat balance:

Heat supplied − heat leakage = heat flow through glass
61.1 − 27.8 = 33.3 Btu per (hr) (sq ft)
which checks 35.1 determined by direct computation within 5 percent.

APPENDIX B

ESTIMATION OF NET RADIANT ENERGY EXCHANGE BETWEEN GLASS AND ATMOSPHERE

Average data for February 23, 1948; 1:30—2:00 p.m.

1. Temperatures, Fahrenheit:

Calorimeter Surface	44.6
Glass surfaces (assumed no drop through glass)	39.4
Outdoor Air, dry bulb	46.0
Outdoor air, dew point (approx)	19.0

2. Miscellaneous:

Wind velocity	6.8 mph from east
Sky	Cloudless with slight haze
Position of glass	6 deg off horizontal
Total solar intensity	160 Btu per (hr) (sq ft)
Incident angle of sun	60 degrees

Computations

1. Heat gain by convection from air:

   With parallel flow of air, it is assumed that the convection coefficient for standard air, a surface length of 6.17 feet (width of glass plus entrance length) and Y/L of 0.60 (see Reference 7, Equation 6, p. 27 and Fig. 13, p. 29) is 1.88 Btu per (hr) (sq ft) (F deg) for the 6.8 mph wind. A 3 percent correction for density makes this 1.94.

   Therefore

   $q = 1.94\ (46.0 - 39.4) = 12.8$ Btu per (hr) (sq ft)

2. Heat gain by convection from calorimeter ..... negligible

3. Heat gain by radiation from calorimeter, Btu per (hr) (sq ft) from Equations 4 and 5 ..... 4.2

4. Absorbed solar energy, Btu per (hr) (sq ft)
   160×0.02 ..... 3.2

5. Radiation exchange with ground surroundings ..... negligible

6. Heat lost by glass in radiant energy exchange with sky, Btu per (hr) (sq ft)
   $12.8 + 4.2 + 3.2$ 20.0

Comparison with Brunt's Empirical Formula

1. Incoming long-wave length radiation from atmosphere, Btu per (hr) (sq ft)

$$0.173\left(\frac{460 + 46}{100}\right)^4 (0.44 + 0.47\sqrt{0.098}) \quad \ldots\ldots 66.6$$

2. Outgoing radiation from glass, Btu per (hr) (sq ft)

$$0.937 \times 0.173\left(\frac{460 + 39.4}{100}\right)^4 \quad \ldots\ldots 100.5$$

3. Net radiation loss, Btu per (hr) (sq ft)
   $100.5 - 66.6$ 33.9

APPENDIX C

PROCEDURE IN COMPUTING THE INFLUENCE OF CEILING RADIANT PANEL TEMPERATURE ON THE OVERALL COEFFICIENT

1. Conditions listed on Fig. 4 have been arbitrarily selected. The shape factor, glass-to-ceiling of 0.30 represents a 10 x 6 ft high window in the center of one wall of a 20 x 20 x 10 ft room. The shape factor of the glass to the rest of the room is therefore 0.70.

2. Equations were set up, one for the heat flow to the glass from the indoor surroundings, the second for the heat flow from the glass to the outdoor surroundings. In each equation heat flow was a function of glass temperature. Therefore, a plot of these two equations for a given ceiling panel temperature and a selected expression

for convection heat transfer, indoor air-to-glass, resulted in an intersection which gave at once the unknown glass temperature and the heat flow quantity.

3. The radiation exchange between glass and outdoor ground surroundings was lumped with the convection coefficient.
4. The value of U was in all cases determined by dividing heat flow by the air-to-air temperature difference, 60 deg.

DISCUSSION

C. S. LEOPOLD, Philadelphia, Pa. (WRITTEN): This paper again underscores the importance of a separate analysis for radiation and convection. The serious error in utilizing the combined coefficient was treated at some length in my two papers on mechanism of heat transfer and panel cooling, presented to the *A.S.R.E.* In these papers, it was shown that, in picking up a portion of the energy from a high-temperature source, such as a luminaire, by means of a cooled surface, the combined radiation and convection coefficient as frequently, though erroneously, used could be made to vary from a low value of approximately 1.5 to a high value of infinity, or even be made negative.

The assumption that, with small temperature differences, the error in the combined coefficient is likewise small is not in accordance with fact and the deviation is of engineering significance as has been indicated in this paper by such basic considerations as the performance of ordinary windows.

When an attempt is made to understand the actual performance of heating or cooling systems, the error in using the combined surface-conductance concept substantially precludes an understanding of the problem. Radiation, of course, may occur between a gas and a solid or solid to solid. Where the latter is the major consideration—as, for example, the radiation between the inside structure and the glass of a window—radiation from the structure to the window presupposes a change in the temperature of the structure surface unless some means is provided to restore the energy loss from the structure to the glass by radiation. If there is a change in the surface temperature of structure, there will be a resulting temperature change in the depth of the structure; and heat-storage phenomena thereby become important because a minor change in temperature of a structure can cause a significant change in the heat balance. It thereby becomes important to know not only the overall heat transfer through a window, but also its distribution. For example, if we hold inside and outside air temperatures constant, the heat loss from a room heated by warm air would appear to be less than if the heating were accomplished by standing radiation. In the case of air heating, the structure temperature would drop with time, thus decreasing radiation to the window; whereas, with standing radiation, the drop in temperature of the structure would, in part, be counteracted by radiation from the radiator.

H. E. ROBINSON,[13] Washington, D. C. (WRITTEN): Neither the text nor the tables of the report state that the glass under investigation is placed in a vertical plane.

The use of a U-value based on the inside surface (calorimeter) temperature is objectionable in a report issued by the A.S.H.V.E. A U-value, by definition accepted by the Society, indicates a heat transfer coefficient based on the temperature difference from air to air. Further, such a 'U-value' is not necessary, since air temperatures at 18 in. from the glass are available and those published in Tables 1 and 2 appear to be consistent with the heat flow rates; *i.e.*, the air is 0.048 deg cooler than the calorimeter surface for each Btu per (hr) (sq ft) of heat flow rate.

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[13] Mechanical Engineer, U. S. Department of Commerce, *National Bureau of Standards*, Washington 25, D. C.

Although in tests described in Reference 1 thermocouples were cemented in grooves in the inner face of the glass, this paper mentions only the use of thermocouple probe on page 3. Since there is some doubt in the writer's mind as to the reliability of measurements with such a probe, he would like to know the degree of agreement between measurements by the two methods.

I would like to know how well the shielded thermocouples used for measuring the air temperature inside the calorimeter 18 in. from the glass agrees with fine wire unshielded thermocouples in the same position, particularly when the calorimeter was receiving solar radiation.

The use of the term *nocturnal radiation* is unfortunate, since it suggests a phenomenon occurring only at night, although the attempt to measure it, summarized in Table 4, is based on data taken during the day. We would suggest the use of a more general term, such as *sky radiation*. In this connection, it may be remarked that, during clear weather, loss of heat by radiation to the sky may tend to sub-cool a surface below the surrounding air temperature, but that, during the day, diffuse or indirect solar radiation from the sky may offset or reverse this effect. It is noted that in Appendix A, no entry is made to account for sky radiation, and that, nevertheless, the heat balance checks within 5 percent with the directly computed heat loss, which indicates little or no net sky radiation effect.

The magnitude of the sub-cooling of the glass surface below the outdoor air temperature reported in the 2nd and 3rd lines of Table 4 seems surprisingly great, and gives rise to the question as to how and where the outdoor air temperature was measured and whether the measurement reported did not include some radiation effect. In connection with Appendix B where these data are computed, it is noted that the radiation from the calorimeter to the glass is 4.2 Btu per (hr) (sq ft) but that convection from the calorimeter to the glass is stated to be negligible. I would expect radiation and convection from the calorimeter to the glass to be of approximately the same magnitude, particularly since the direction of heat flow is upward.

In the text and in Fig. 4, the authors refer to low indoor and moderate indoor convection effects. I do not understand under what circumstances such different effects might be obtained. The convection from the indoor air to the indoor surface of the glass will vary in accordance with a function such as that quoted from Griffith & Davis (Reference 5). The magnitudes of the coefficient and exponent cited may or may not be correct, but presumably there are some approximately constant values which will pertain for the range of conditions considered. It may be necessary to determine these values by further research, but it is felt that a wrong implication is given when the authors present curves which suggest that there may be two or more levels of convection effect for the same conditions. Possibly the authors mean that the two curves bracket the correct curve, but, if this is so, the authors should not use the data of Fig. 3 for establishing the lower curve. Fig. 3 involves the temperature difference between the calorimeter surface and the glass and the convection effect considered is necessarily between indoor air and glass, so that the coefficients of Fig. 3 are necessarily materially lower than they should be. I would not favor the inclusion in the report of Fig. 4 as it stands at present.

I agree with item 4 of the summary of the paper to the effect that a U-value of 0.45 for double-glazed windows is too low, and also with the substance of item 3 of the summary. I feel, however, that the paper would be improved if the exposition of the problem of the exchange of heat to and from window glass and surroundings by radiation in parallel with that by convection were made in a more clear-cut manner. For example, I would like to see a paper in which the heat transfer coefficient for convection from (still) indoor air to the glass was definitely established over the range of conditions probable in application; also similar data for the coefficient for convection from the outdoor surface of the glass to the atmosphere under varying conditions of wind intensity and direction. The method of computing the

radiation transfer from various indoor surfaces to the glass should be more fully presented than it is in this paper. Radiation from the glass to the surroundings and to the sky should be given more precise definition and evaluation. It is agreed that more satisfactory data on the sky radiation is required before a complete paper on the subject can be written. To bring the entire paper into focus, a complete example representing a typical case should then be worked out making use of the separate and individual coefficients outlined.

C. M. ASHLEY, Syracuse, N. Y.: In conversation and in correspondence with the author, the interesting point was developed that the double glass figures which we have been using all along were based upon a misconception as to the correct method of calculating the heat transfer. I would strongly urge that we take the necessary steps to correct the values for double glass promptly. It is obvious that they are not correct, both from the theoretical examination and from the practical results of this paper, which are most significant.

I would also like to emphasize what Mr. Leopold has said. It is quite improper to assume the same inside wall temperature as glass temperature. The effect of this upon the heat transmission through the glass may be quite considerable. If one assumed that all of the surface of the room were made of glass, one would have only the convected portion of the transmission. It would, I believe, be something less than half of the present calculated amounts. Actually, of course, that portion of the heat which comes in by radiation has to go first to other surfaces and then, by convection, from the other surfaces to the air. In that process, it is inevitable that there will be a lower temperature difference between the other surfaces and the air than between the glass surfaces and the air. Because of this, the single glass factor, as it is actually used, is undoubtedly too high and I believe we should put forth considerable further effort on that part of the problem and try to systematize our understanding with relation to specific room structures.

CYRIL TASKER, Cleveland, Ohio: This study is part of a long-range program being conducted at the Laboratory. Professor Tuve recently announced that Research Bulletin No. 1,[14] covering a study on heat transmission through glass that was made by Mr. Parmelee as a first step in this long-range program, was available to all members of the Society.

We have been encouraged by the number of letters received from members requesting this bulletin. One of the chief functions that the Research Laboratory or the Committee on Research can perform for the membership is to review the literature on a particular subject or in any particular field of interest, analyze it critically and prepare a summation of what has been done before, and what is still unknown, or remains in doubt. Bulletin No. 1 does that exactly.

It is an excellent piece of work by Mr. Parmelee and worth having. Council has authorized the furnishing of one free copy to each member, upon application. You may obtain yours by writing to Society Headquarters, 51 Madison Ave., New York 10, New York.

R. A. MILLER, Pittsburgh, Pa.: Mr. Ashley presented the thought that we need more data on values of wall radiation, loss through window and from walls, the loss by conductance and convection from the air, etc.

The Technical Advisory Committee on Glass will be most grateful for any suggestions regarding how to determine some of those things which members want to know. The development of this calorimeter is quite an achievement. We realize that there are many things that it cannot do. We should like to know how to modify them so we can achieve more.

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[14] A.S.H.V.E. RESEARCH BULLETIN No. 1, Vol. 53—Heat Transmission Through Glass, by George V. Parmelee (published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 51 Madison Ave., New York 10, N. Y. Price $2.00 per copy).

The whole foundation of the work of the Technical Advisory Committee on Glass is in Bulletin No. 1, referred to by Mr. Tasker. It was felt that we could not proceed actively and properly to continue to develop research on heat transmission through glass without having a pretty complete knowledge of what had been done previously and without getting some kind of a correlation of these data before attempting to improve upon it.

The value to the Committee and to the persons particularly interested in heat gain and loss to buildings, we feel, has been very much more than the possible cost of such research. We hope that you will all help in carrying on the research in such a fashion that members will derive the benefits which should result from such an investigation.

AUTHORS' CLOSURE: I am glad that Mr. Leopold emphasized the weakness of the overall coefficient concept as a means of calculating heat flow. It is a convenient method, but, in many cases, an inaccurate one, since the mechanisms of heat transfer by convection and by radiation are quite unrelated. This concept also fails when a structure is not in thermal equilibrium. We have already had to abandon the overall coefficient in computing the heat flow through sunlit walls and roofs. As shown in the paper, the exchange of radiant energy and the flow of heat by convection may often be in opposite directions. The temperature difference between a heat-transfer surface, such as a window, and the ambient air cannot alone indicate whether or not such a condition exists. Mr. Robinson's excellent comments on the paper are appreciated. With regard to the position of the glass, the data are for tests made with the glass vertical. In a few cases, not reported, tests were made with the glass horizontal, or inclined at an angle.

With regard to basing the U value on the inside surface (calorimeter) temperature, we can only say that it was impossible to select an indoor air temperature, as this temperature was dependent upon the distance from the glass. We found that shielded couples four inches from the glass gave temperatures from one to three degrees lower than air temperature couples 18 in. from the glass. Hence, the U values would have varied, depending upon which air temperature measurement was used. Also, in an enclosed air space, the convection heat transfer is generally considered to be dependent upon the temperatures of the boundary surfaces. The surface temperature, therefore, was considered to be a logical reference point in computing the U value. The conventional use of a U value, when radiative components are present, is, at best, an approximation, since air-to-air temperature difference is no indication of the radiant heat exchanges that take place. A practical solution is to give heat flow rates for specific indoor and outdoor conditions.

The thermocouple probe temperatures were roughly a degree higher than the corresponding glass temperatures indicated by embedded thermocouples. Therefore, they indicate relative temperatures. Evidently, the felt backing pad insulated the glass area to some degree, although the contact time was short.

We did not use any fine wire unshielded couples at the 18 in. location to measure air temperature. Two such couples were used four inches from the glass, but were shielded by two concentric aluminum foil shields. Studies made at the Laboratory several years ago showed that these shields were unsatisfactory in the presence of radiant beams of considerable flux density, such as from the sun or glowing objects.

We agree with Mr. Robinson that the term *nocturnal radiation* is a poor one to describe this phenomenon which occurs day and night. The term *sky radiation* is too general, however, since it does not distinguish between long-wave length and the short-wave length scattered solar radiation. Neither opaque nor transparent substances react the same to these two radiations. The heat balance checks in Appendix A involved only the heat supplied the calorimeter, and, therefore, no entry was necessary to account for either radiation exchange or convection at the outdoor surface of the glass.

The outdoor dry bulb temperature was measured by a shielded and aspirated couple and is believed to be correct. It checked closely with sling psychometer readings which were taken at regular intervals. The depression of the glass temperature below the air temperature as noted in Table 4 has been observed frequently, particularly with the glasses which absorb little solar radiation. In Appendix B, we should, perhaps, have included the quantity of heat convected from the calorimeter to the glass. Fig. 2 will show, however, that this quantity was much less than one Btu per (hr) (sq ft) and, therefore, we consider it as insignificant.

Fig. 4 was presented primarily to indicate the fallacy of basing U values on air-to-air temperature difference when radiative effects are present. It also seemed like a good idea to show how significant the indoor conductance for convection could be. Neither curve is intended for design purposes. It is the authors' opinion that our data on natural convection heat transfer is far from complete for flat surfaces of some size. No two sets of data cited in the literature are in agreement with each other, so it is evident that much laboratory and field work needs to be done on this subject.

A curve identical to Fig. 3, but a function of air-to-surface temperature difference rather than surface-to-surface difference, was used in computing one of the curves of Fig. 4 because it was believed that such a convection function for a wall surface in a room could conceivably exist. It is believed that so-called face convection heat transfer in rooms may be represented by many different curves, depending upon air movement in the enclosure and ventilation rates.

We are in agreement with Mr. Robinson on the point of definitely establishing convection data for both indoors and outdoors and data for the sky radiation. We find that this information is necessary in the problem of evaluating heat gain through sunlit glass. This is the reason for making an investigation to establish the curve shown in Fig. 3. Also, we have recently designed and constructed a convection-compensated radiometer to evaluate the sky or nocturnal radiation exchange, and, by difference, to arrive at convection data for outdoor surfaces. We hope to pin down some of these factors close enough for practical use.

In reply to Mr. Ashley's comments on the data for THE GUIDE, we are looking forward to improving our information for heat loss through glass. I think you will recognize from the points that were made in the paper that there will not be just one value for single glass and one value for double glass. There is more to it than that. In connection with the absence of radiant-energy exchange in an all-glass enclosure, as Mr. Ashley pointed out, hot-box tests have shown that the overall coefficient is materially reduced with a reduction in radiation effects. A number of years ago, tests were made with the hot-box lined with aluminum foil, which is, as you know, highly reflective and has an extremely low emissivity. The overall coefficients resulting from these tests were materially lower than tests made with somewhat similar apparatus, but with so-called black surfaces. You will find the information on these particular hot-box tests in the bulletin mentioned by Mr. Tasker and Mr. Miller.

By Mr. Ashley's concluding statement that *we try to systematize one's understanding with relation to specific room structures*, I take to mean that we should investigate the heat-transfer conditions in actual rooms. This would involve determination of room-surface temperatures and the resulting radiant-energy exchange. It would also include a study of convection heat-transfer conditions under actual field conditions. I am much in favor of making investigations of this type. Mr. Miller has pointed out that, in our calorimeter apparatus, these matters are well established, but that the apparatus does not tell us everything we must know.

1352

No. 1352

THE INTERNAL PERFORMANCE OF AN INDUCED DRAFT, PACKED COOLING TOWER

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the University of California.

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BECAUSE of the complexity of the mechanism of evaporative cooling in packed cooling towers, only overall characteristics have been obtained by the different experimenters [1-6]. The forced draft towers previously employed at the University of California were small and were placed indoors to insure steady ambient conditions. A larger tower was therefore constructed to facilitate more extensive and detailed study and attempts were made to equip the tower with instruments for obtaining information on the internal mechanism of cooling. The objective of the tests was to obtain the following items:

1. A determination of point enthalpy transfer coefficient per unit volume of tower $(h_1a)_x$, as a function of heights in the packed section for various water rates and a constant air rate.

2. A comparison of the overall enthalpy transfer coefficient per unit volume of tower with its components due to the cooling of water drops, packing water films and wall water films.

3. A comparison of the overall enthalpy transfer coefficient per unit volume of tower (including cooling in the spray section and packed section), $(h_1a)_T$, and its components due to spray only, $(h_1a)_s$, and packed section only, $(h_1a)_p$.

4. A determination of the variation of the temperature of the water in the packed section as a function of the tower cross section and height for different water rates and a constant air rate.

5. A determination of the variation in mixed mean temperature of the water in the packing, on the walls and of the total water at different tower heights for various water rates and a constant air rate.

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[1] Exponent numerals refer to References.

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6. A determination of the distribution of water in the packed section in terms of the percentage of water film on the packing surfaces, water drops between packing slats and water film on the tower wall surface.

7. A determination of the pressure drop through the packed section for different water rates and a constant air rate.

Three groups of tests were run at approximately the *same air rate*. Some of the items in the foregoing objectives were not fulfilled because of insufficient data and are only briefly mentioned.

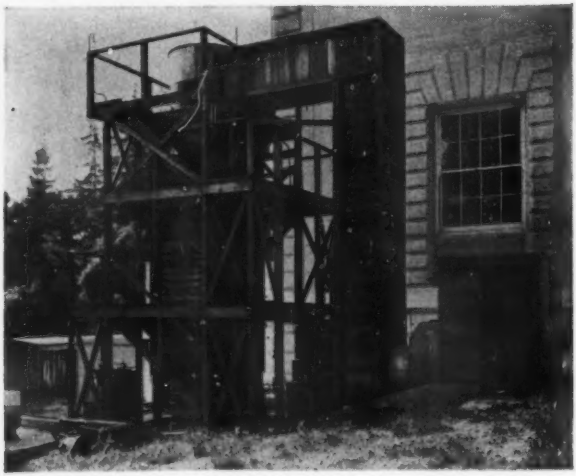


FIG. 1 EXPERIMENTAL COOLING TOWER

DESCRIPTION OF APPARATUS AND TESTS

Tower: The cooling tower was a mechanically induced draft, counterflow, packed or spray type tower. Fig. 1 shows the general arrangement of equipment for the tests and Fig. 2 shows a schematic diagram of the experimental arrangement.

Briefly, the cooling tower consisted of a vertical rectangular redwood tower 57 x 61 in. with double walled 1 in. x 6 in. wood siding which was removable on two opposite sides in 3-ft sections. The packed section was 152 in. high, with 33 decks of redwood packing, each of which consisted of 27 or 28 horizontal slats ¼ in. (base) x 4⅛ in. (height) x 57 in. (length). The slats were staggered in alternate decks and the decks were mounted on 4 dowel pegs for ease of replacement. Below the packed section, air was admitted by means of four sets of louvers 2 ft high, one set on each side of the tower with each set shaped to direct the incoming air to the packed section without permitting loss of falling water from the system. Water circulation during a run was maintained in a closed system. A catch basin was placed at the bottom of the tower and con-

nected to a sump tank which held approximately 1320 gal. The sump tank area was small at the normal water level in order to exaggerate the changes in observed water level due to losses from the system by evaporation or entrainment. A suction line connected to the bottom of the tank led the cooled-water to a circulating pump at ground level at one side of the tower. A metering orifice was placed between the pump and the shell-and-tube heat exchanger, which was heated by steam. From the exchanger, the water was distributed to nine 1½ in. spray nozzles located 15 in. above the packing.

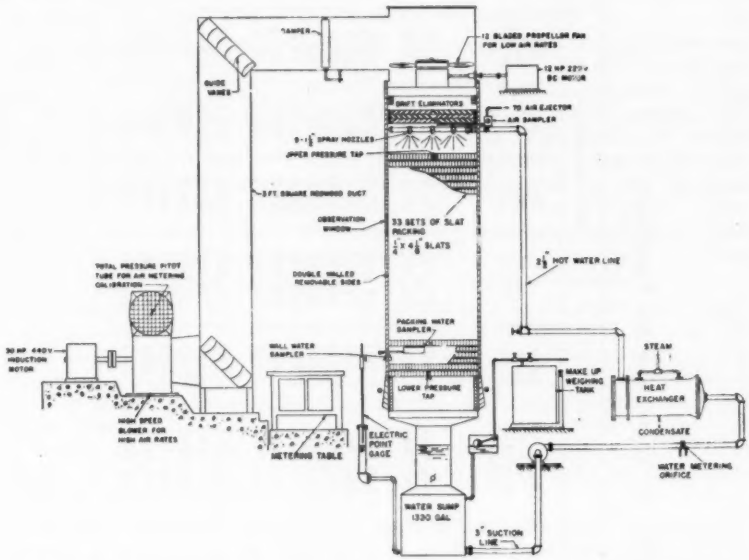


FIG. 2. SCHEMATIC DIAGRAM OF THE EXPERIMENTAL COOLING TOWER AND INSTRUMENTATION

Water which was evaporated or entrained into the outgoing air was automatically replaced from a weighing tank, the flow of make-up water being controlled by a float valve to maintain a constant water level in the sump tank (see Fig. 2).

During the first two tests, air flow was established by means of a propeller fan located in the fan nozzle at the top of the tower and driven by a variable speed d-c motor. In the third test a duct was attached to the fan nozzle and connected to a high speed centrifugal fan. As shown by Fig. 2, air flow could be established by either the propeller fan or the centrifugal fan. The propeller fan was to be used for low air rates and the centrifugal for high air rates. It was found that the high speed centrifugal fan was unsatisfactory at low air rates due to pulsating flow when near the shut-off point. In future tests it will be only used for high air rates.

TABLE 1. WATER AND AIR CONDITIONS

WATER CONDITIONS							AIR CONDITIONS IN				AIR CONDITIONS OUT		
Data Set Number	Run Number	Water Rate Lb/hr L	Temp in F T_1	Temp out F T_2	Heat Loss Btu/hr $\Delta T x L$	Evap Rate Lb/hr e	Air Rate Lb/hr G	Dry Bulb F t_2	Wet Bulb F t_{wb2}	Enthalpy Btu/lb i_{a2}	Dry Bulb t_1	Wet Bulb F t_{wb1}	Enthalpy Btu/lb i_{a1}
I	1	92,100	101.3	81.3	1,842,000	1,000	75,700	59.2	56.0	24.0	88.0	84.6	48.3
	2	70,100	104.4	76.4	1,989,000	1,300	76,500	58.7	56.0	24.0	87.5	85.7	50.0
II	1	95,500	105.6	87.4	1,740,000	1,360	61,500	57.5	55.6	23.6	88.2	87.1	51.9
	2	63,500	114.9	82.7	2,044,000	1,420	71,300	58.5	56.5	24.3	89.1	88.0	53.0
	3	42,000	125.0	78.4	1,960,000	1,550	59,400	57.0	55.6	23.6	92.0	91.3	56.6
	4	32,500	118.5	74.4	1,434,000	1,240	63,400	59.8	56.9	24.4	83.1	83.1	47.0
	5	24,700	129.9	72.0	1,431,000	1,230	68,100	59.6	57.0	24.6	82.0	81.9	45.6
	6	13,300	94.6	65.6	386,000	470	65,400	58.9	56.4	24.2	65.0	65.0	30.1
	7	6,500	121.6	58.8	408,000	390	80,000	59.1	56.2	24.1	63.8	63.8	29.2
III	5	73,000	85.7	64.4	1,555,000	1,290	78,300	46.2	41.4	15.9	65.4	65.4	35.8
	6	48,500	85.3	61.0	1,180,000	1,220	79,000	44.9	41.8	16.1	62.2	61.2	31.0
	1	43,000	87.0	60.5	1,140,000	830	74,000	48.8	45.3	18.0	66.0	66.0	33.4
	2	31,500	86.4	56.9	928,000	920	72,700	46.6	42.2	15.8	61.0	61.0	28.6
	3	22,300	86.4	55.4	691,000	800	81,700	49.2	43.6	17.0	58.3	57.2	25.5
	4	11,500	85.8	51.2	398,000	510	85,000	49.2	42.6	16.5	54.2	52.7	21.2

Measurements and Measuring Instruments: For the first two test groups, air rates were obtained by making a heat balance between the gain in enthalpy of the air and the loss in enthalpy of the known rate of water. An air rate for the third test group was obtained by drawing the air by means of a fan from the top of the tower through a 3-ft square wooden duct and a calibrated metering nozzle in which its velocity was determined. A calibrated orifice downstream from the centrifugal pump was used to meter the water.

Evaporation and entrainment rates were determined from the weighed make-up water plus the loss indicated by a change in the water level in the sump tank. An electric point gage was used to measure the water level.

Inlet water temperatures were obtained by means of four thermocouples connected in series and located in the spray nozzle manifold. A thermocouple was located in the sump tank in order to obtain exit water temperatures. Seven movable thermocouples were located from the bottom to the top packing deck in order that a traverse of water temperature with height and cross section could be obtained. It was realized that the temperature distribution of the water over the cross section might vary in different directions, but the best that could be done at the time was to obtain a traverse along only one center line. Thermocouple emf's were obtained by means of a potentiometer readable to 0.005 mv. The potentiometer was mounted on a metering table placed upon a concrete slab away from the cooling tower in order to eliminate vibrations.

The pressure drop in the tower was measured with an electric point gage micromanometer readable to 0.0002 in. of water. This instrument was also placed on the metering table. The pressure taps were located at either end of the packed section and in the center of the packing deck. The tap consisted of two concentric cylinders with the annulus closed on the ends. Holes were drilled on the inner cylinder and a pressure tap was led from the outer cylinder wall to a water trap and then to the manometer. Another line was led from the bottom of the outer cylinder wall to the water trap in order to drain water caught in the pressure tap annulus.

Inlet air wet bulb and dry bulb temperatures were determined by four thermocouple psychrometers, one located at the center of each intake louver. Readings were taken separately and averaged at a later time. An air sampler was used to determine outlet air wet bulb and dry bulb temperatures just beyond the sprays. A thermocouple psychrometer located downstream from the propeller fan was used as a check.

The air sampler was constructed from two tinned cans, one within the other. Air from the tower was led into the outer can over a baffle, acting as an entrained-water scrubber, then into the inner can over the wet and dry thermocouples. Air from the inner can was evacuated by means of an air ejector. The outer can was thoroughly insulated. This method yielded a measurement at only one point in the cross section. It is realized that the non-uniformity of the temperature of water would cause some non-uniformity of the exit air conditions. The exit air conditions, as measured, were not considered to be as accurate as desired.

The distribution of the total water in the tower between the falling water drops, the film on the packing and the film on the tower walls was determined by means of a packing water catcher and a wall water catcher. The packing water catcher was shaped into a long rectangular trough with a false bottom.

The lower chamber was divided longitudinally into two enclosures. The upper chamber was open at the top but was partitioned crosswise by means of bulkheads. These bulkheads were spaced in such a manner as to catch packing water in one space and water drops in the other when placed just beneath a packing deck. By punching holes in the false bottom partition in such a way as to lead only water drops between the packing slats into one of the lower longitudinal chambers and slat film water into the other, it was possible to collect each of these separately and conduct the water to the outside of the tower by means of two rubber hoses connected to each of the lower chambers. The

TABLE 2. WATER DISTRIBUTION

DATA SET NUMBER	RUN NUMBER	FILMS				DROPS				WATER TO AIR RATIO
		On Wall Lb/hr W_w	% of Total % W_w	On Slats Lb/hr W_s	% of Total % W_s	Between Slats Lb/hr W_d	% of Total % W_d	+ Films Lb/hr W_s+W_d	% of Total % (W_s+W_d)	L/G
I	1	31,500	38.9	27,000	24.3	33,600	36.8	60,600	61.1	1.22
	2	27,300	38.9	17,100	24.4	25,800	36.7	42,900	61.1	0.92
II	1	42,000	44	25,800	27	27,700	29	53,500	56	1.55
	2	27,900	44	17,800	28	17,800	28	35,600	56	0.89
	3	17,200	41	13,400	32	11,400	27	24,800	59	0.71
	4	11,700	36	12,300	38	8,500	26	20,800	64	0.51
	5	8,100	33	10,400	42	6,200	25	16,600	67	0.36
	6	5,450	41	4,000	30	3,850	29	7,850	59	0.20
	7	3,120	48	1,430	22	1,950	30	3,380	52	0.08
III	5	58,500	19.9	23,900	32.7	34,600	47.4	14,500	80.1	0.93
	6	37,400	22.8	15,600	32.2	21,800	45.0	11,100	77.2	0.62
	1	32,000	25.7	13,500	31.5	18,500	42.8	11,000	74.3	0.58
	2	21,400	32.0	9,300	29.5	12,100	38.5	10,100	68.0	0.43
	3	14,000	37.1	6,500	29.2	7,500	33.7	8,300	62.9	0.27
	4	8,100	29.2	3,700	32.5	4,400	38.3	3,400	70.8	0.14

wall water catcher consisted merely of a flat trough, fastened to the tower, and draining through a pipe to the outside of the tower.

DATA AND COMPUTATIONS

Evaporated and Entrained Water: The measurement of make-up water by means of the make-up weighing tank and sump tank level yielded the sum of the evaporated and entrained water. Evaporated water could have been obtained only if it were possible to measure the entrained water. Because this measurement was not possible, the total make-up water was construed to be equal to the evaporated water (See Tables 1 and 2). A small amount of entrained water was observed leaving the tower. It was also noted that the entrainment increased sharply at an air flow rate of approximately 5000 lb per (hour) (square foot of net cross section area of packing). For an 8 millimeter water drop formed at the bottom of a slat, the corresponding Reynolds number would be approximately 2800. According to Nottage and Boelter,[7] a Reynolds number of about 4000 is needed to suspend the drop. For the smaller drops, the pre-

viously mentioned air loading (air flow rate) would entrain drops smaller than 2 millimeters.

Water Distribution: The water draining off the packing as film, that falling between packing in drops, or that draining off the wall as film was collected separately to obtain the rate per hour for each (See Table 2). The total drained off the slats as film was obtained by multiplying the weight collected from a portion of the slats by the ratio of total slat length to this portion.

It was found that the sum of the three water quantities obtained in this manner was greater than the quantity determined by means of the metering orifice. Each of the three quantities was therefore decreased in the same ratio so that the total equaled the metered quantity.

Air Enthalpies: Air enthalpies at various heights of the tower were obtained from the actual wet and dry bulb temperatures. The measurement of wet bulb and dry bulb temperatures of a sample of air which is surrounded by water drops requires that the measuring device be protected from these drops of water. When a sample is obtained by passing the air through a tube to the outside and then through a separator in order that any entrained water drops be removed, it is evident that the walls of the device would be wetted and thus cause a change in enthalpy of the sampled air. Because of the small amounts of water involved, any correction would be difficult to obtain. An instrument for making an accurate measurement of the air condition inside the tower has not yet been devised.

Measurements which could readily be obtained inside the tower were the water temperatures; of these, a traverse was obtained along a center line of the packing at various heights in the towers. Wall water temperatures were also measured at the same height. By using the previously measured water distribution of wall water and packing water, a *mixed mean temperature* at each level was computed. The temperature of the water which fell into the basin was also measured. By knowing the total water rate and temperature differences between successive levels in the tower, an enthalpy change was calculated between these successive positions from the sump to the nozzles. Since the wet and dry bulb temperatures of the incoming air were also measured, successive values of the air enthalpy were obtained by equating the changes in water enthalpy to the changes in air enthalpy. The changes in enthalpy between these stations were weighted so that the sum of these increments equaled the total change in enthalpy throughout the tower (See Tables 3, 4 and 5).

Enthalpy Transfer Coefficients: With the mean water temperature and the air enthalpies at various heights in the tower calculated, a transfer coefficient was obtained between successive heights (See Table 6). An energy transfer equation between two successive stations in the tower may be written as,

$$L\, C_{pw}\, (\Delta t)_x = (h_{ia})_x\, V_x\, (\Delta i_x)_m \quad \ldots\ldots\ldots\ldots \quad (1)$$

where

L = water rate, pounds per hour.

C_{pw} = specific heat of water, Btu per (pound) (Fahrenheit degrees).

$(\Delta t)_x$ = temperature increment between two successive heights in the tower, Fahrenheit degrees.

x = height above the bottom of the packed section to halfway between the successive stations considered, feet.

TABLE 3. TOWER WATER TEMPERATURES IN RELATION TO HEIGHT

Data Set Number	Run Number	Average Water Temperature Across Packing From Traverses—As A Function of Height x, Fahrenheit Degree							Mixed Mean Temperature of Packing Plus Wall Water As A Function of Height x, Fahrenheit Degree						
		x=O ft	2	4	6.6	9	10.7	12.6	x=O ft	2	4	6.6	9	10.7	12.6
I	1	75.5	84.9	89.3	92.8	94.6	96.0	97.7	80.2	87.2	90.8	93.7	95.6	97.0	99.0
	2	77.8	80.4	84.0	90.6	93.7	95.5	96.0	83.2	85.6	88.6	93.5	96.1	98.1	99.2
II	1	83.2	85.5	89.4	96.5	98.3	98.9	99.8	88.0	91.7	93.2	98.5	100.1	100.8	101.8
	2	74.2	77.3	82.6	89.8	95.3	96.5	98.8	82.5	84.9	88.7	94.6	99.4	100.7	103.3
	3	71.5	74.2	78.3	87.1	95.5	99.3	104.8	81.3	83.8	86.7	95.0	102.3	105.2	109.1
	4	70.8	73.8	78.6	84.7	94.7	100.0	102.2	79.1	81.9	85.2	91.7	97.9	103.5	105.7
	5	68.2	73.8	74.2	89.3	96.5	101.3	105.5	78.7	82.6	88.0	96.2	102.8	107.0	110.1
	6	59.3	60.7	63.3	65.3	70.8	74.3	80.7	64.8	66.0	68.3	71.6	76.5	79.5	85.0
	7	55.6	55.8	56.5	58.5	62.0	66.0	79.2	64.8	65.2	67.2	72.2	78.5	83.4	92.0
III	5	58.9	61.9	63.6	70.5	74.5	76.7	78.4	71.4	72.2	73.6	77.5	80.7	81.8	82.5
	6	55.5	56.8	61.2	68.7	72.1	73.1	76.3	69.3	69.8	71.5	76.2	79.2	80.5	81.5
	1	51.0	52.7	55.8	63.5	68.4	70.1	73.5	65.5	66.1	67.8	72.8	76.6	78.2	79.5
	2	45.0	45.9	47.5	53.8	58.7	60.6	69.3	58.7	59.3	61.0	66.6	72.3	74.4	75.8
	3	60.3	62.7	65.3	72.9	75.4	76.3	77.8	69.3	69.8	71.5	76.2	79.2	80.5	81.5
	4	56.7	57.4	61.7	68.8	72.1	72.9	75.5	67.1	67.8	70.0	74.5	77.3	78.5	79.5

TABLE 4. WATER TEMPERATURE DISTRIBUTION ON WALL AS FUNCTION OF HEIGHT x

DATA SET NUMBER	RUN NUMBER	FAHRENHEIT DEGREE						
		x=0 ft	2	4	6.6	10.7	9	12.6
I	1	89.3	91.5	93.7	95.5	99.0	97.3	101.3
	2	91.1	92.9	95.1	97.3	101.3	99.1	103.5
II	1	94.2	97.3	98.2	100.9	103.0	102.2	104.0
	2	92.8	94.6	96.4	100.9	105.6	104.4	107.5
	3	95.0	97.7	99.1	106.6	114.2	112.0	115.8
	4	94.2	95.5	96.8	102.2	109.8	107.5	111.5
	5	97.3	100.4	101.3	109.3	118.5	115.5	119.7
	6	72.9	73.8	75.5	80.9	87.1	84.9	89.3
	7	75.1	75.5	78.7	87.1	102.2	96.4	106.2
III	5	73.7	75.1	75.5	79.1	82.7	82.2	83.5
	6	72.9	73.3	74.2	78.2	82.2	81.3	83.1
	1	70.2	70.6	72.0	76.4	80.9	80.0	81.7
	2	65.3	66.2	67.5	72.9	80.5	79.1	82.2
	3	73.3	74.2	75.1	78.2	82.6	82.6	83.1
	4	70.6	71.6	73.3	76.9	80.5	79.5	81.3

TABLE 5. AIR ENTHALPY VARIATION

DATA SET NUMBER	RUN NUMBER	AIR ENTHALPY AT DIFFERENT HEIGHTS ABOVE BOTTOM OF PACKED SECTION i_{ax} in Btu/lb						
		x=0 ft	2	4	6.6	9	10.7	12.6
I	1	22.6	31.2	35.5	39.1	41.4	43.1	45.5
	2	30.4	32.4	35.5	39.9	42.2	44.1	45.1
II	1	24.5	30.2	32.5	40.7	43.2	44.3	45.9
	2	24.3	26.3	29.7	34.9	39.2	40.4	42.7
	3	25.7	27.5	29.6	35.4	40.6	42.7	45.4
	4	26.8	28.2	29.9	33.2	36.4	39.3	40.4
	5	27.0	28.4	30.3	33.3	35.7	37.2	38.4
	6	24.2	24.8	24.8	25.5	26.5	27.1	28.2
	7	24.6	24.6	24.8	25.2	25.7	26.1	26.8
III	5	22.2	22.9	24.2	27.8	30.8	31.9	32.7
	6	21.2	21.5	22.5	25.4	27.2	28.0	28.6
	1	20.9	21.2	22.2	25.1	27.3	28.2	29.0
	2	16.6	16.9	17.6	20.0	22.5	23.4	24.0
	3	20.8	20.9	21.4	22.7	23.5	23.9	24.2
	4	18.6	18.7	19.0	19.6	20.0	20.2	20.3

TABLE 6. ENTHALPY TRANSFER COEFFICIENTS PER UNIT VOLUME

DATA SET NUMBER	RUN NUMBER	OVER ALL	SPRAY SECTION	PACKED SECTION	POINT VALUES AT DIFFERENT HEIGHTS ABOVE BOTTOM OF PACKED SECTION					
		Btu / hr Ft³(Btu/lb)			$(h_i a)_x$, Btu / hr Ft³(Btu/lb)					
		$(h_i a)_{s+p}$	$(h_i a)_s$	$(h_i a)_p$	x = 1 ft	3	5.3	7.8	9.85	11.65
I	1		244	249	627	333	203	132	160	161
	2	235	374	184	264	267	290	326	161	67
II	1	198	340	154	261	107	291	96	62	73
	2	198	1340	157	130	179	197	183	68	106
	3	157	309	129	110	121	224	175	87	86
	4	108	216	108	113	111	151	115	130	38
	5	112	175	81	119	119	111	81	60	36
	6	81	135	73	18	82	78	100	72	90
	7	59	92	40	10	43	164	138	45	47
III	5	297	470	203	91	161	350	314	157	81
	6	216	306	138	41	141	279	173	105	65
	1	239	597	172	55	148	324	221	131	87
	2	183	552	153	43	118	271	254	123	64
	3	120	139	55	18	60	112	66	39	24
	4	68	84	27	12	37	50	29	17	11

$(h_i a)_x$ = point enthalpy transfer coefficient per unit volume of tower, Btu per (hour) (cubic foot) (Btu per pound of dry air).

a = surface area through which energy is transferred per unit volume in the volume included by $(\Delta t)_x$, square feet per cubic foot.

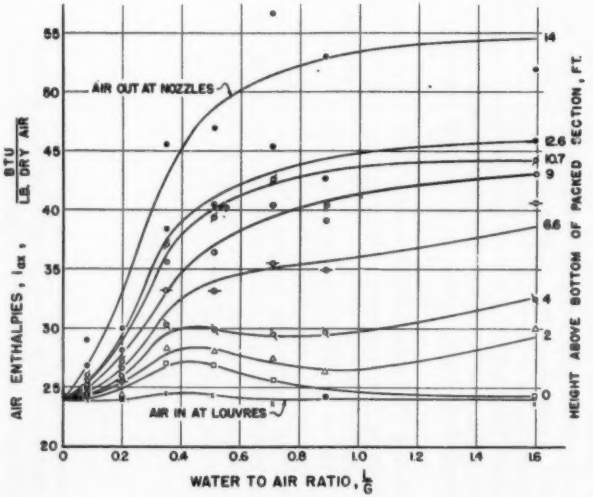
V_x = volume of tower included between the two successive temperature stations considered, cubic feet.

$(\Delta i_x)_m$ = integrated mean enthalpy potential difference[4] between the two successive temperature stations in tower, Btu per pound of dry air.

Note that a was not separated from h_i even though a set of packing was used for which the surface area was known. It will be shown that a considerable portion of the falling water in the tower was in the form of an aggregation of various sized drops for which the surface area was not known. This fact is true for all packed cooling towers where there is an interchange between film and drop formation. Further study on the effect of the percentage of water in the drop form and that in the form of films on unit volume enthalpy transfer coefficients was considered needed. Work is now being carried out by the author to fulfill this need.

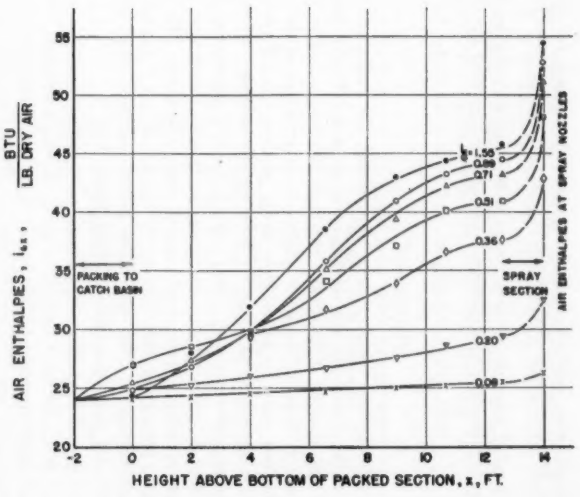
Along with the calculation of a point transfer coefficient was the determination of an overall transfer coefficient for the whole packed section and the spray section only. With the enthalpy changes of the water for the packed section and the spray section known, the following quantities were found in the same manner as indicated by Equation 1:

$(h_i a)_p$ = overall enthalpy transfer coefficient per unit volume for the packed section, Btu per (hour) (cubic foot) (Btu per pound of dry air).



Air rate = 67,000 (± 19 percent) lb per hr.
Data from Test Group 2.

FIG. 3. VARIATION OF AIR ENTHALPY WITH WATER-TO-AIR RATIO AT VARIOUS HEIGHTS



Air rate = 67,000 (± 19 percent) lb per hr.
Data from FIG. 3.

FIG. 4. RELATION OF AIR ENTHALPY TO HEIGHT FOR VARIOUS RATIOS OF L/G

$(h_t a)_s$ = overall enthalpy transfer coefficient per unit volume for the spray section, Btu per (hour) (cubic foot) (Btu per pound of dry air).

Pressure Drop: A small amount of air pressure drop data was obtained for the packed section. No correlation was attempted due to the fact that too few data were taken.

DISCUSSION

Energy Distribution: It was noted that a plot of the air enthalpy, i_{ax}, as a function of height in the tower, x, and of the ratio of water to air rate, L/G, served to represent the energy distribution through the tower better than the temperature of the water. During the tests, values of air enthalpy at the entrance to the tower, i_{a_2}, did not vary by more than three percent for all runs during a given night. A first attempt was made by plotting the enthalpy of the air at a given height in the packed section, i_{ax}, *vs* x with L/G as a parameter, where L was the water rate and G the air rate. In doing so, it was found that some of the experimental points were erratic and caused some confusion. By plotting i_{a_2} *vs* L/G with x as a parameter, the various curves fell into a reasonable order (see Fig. 3 for Test Group 2). By obtaining points on the faired curves of Fig. 3 of i_{ax} *vs* L/G and x, a crossplot was made, see Fig. 4.

It can be noted in Fig. 3 that at an approximate value of $L/G = 0.4$ there was a sharp decrease in i_{ax} for all values of x as L/G was decreased. Although G was maintained at approximately 67,000 lb per hour for these runs, L/G was varied by varying the water rate, L, from 95,500 lb per hour to 6500 lb per hour. It was realized that between the water rates of 95,500 and 0 lb per hour, there would be a transition region from completely wetted to completely dry packing which would be called a partially wetted packing region. In this region there occurs channeling of the water streams on the packing and a consequent loss in cooling effectiveness. The variable L/G was not considered very significant in describing the transition from a fully wetted to a partially wetted packing because G can be changed for a given value of L and still not change relative wetting.

A better variable than L/G for the packing loading would be a quantity which would take into account the water spreading characteristics of the packing. This may perhaps be in part described by the ratio, blocked cross sectional area to the gross sectional area, or by the surface area of the packing for a foot of length of tower per open or net cross sectional area of the tower. In order to prevent the variable from approaching infinity when the packing density approaches that of a spray tower (zero packing surface area), a ratio of slat film water rate to drops water rate was used. This variable to be used in the correlation of results from various towers might be:

$$\frac{L\left(\frac{w_s}{w_d}\right)\left(\frac{A_g - A_n}{A_g}\right)}{S} \quad \ldots\ldots\ldots\ldots (2)$$

where

S = total surface area of packing in tower, square feet.
A_n = net or open cross sectional area of packed section, square feet.
A_g = gross cross sectional area of tower, square feet.

$$\frac{w_s}{w_d} = \frac{\text{water film drained off packing slats, pounds per hour}}{\text{water flow in drops between slats, pounds per hour}}$$

As the density of packing decreases, both area A and the ratio of the film to drop water rate w_s/w_d decrease until the limiting case of a spray tower is reached in which case S and w_s/w_d become zero and

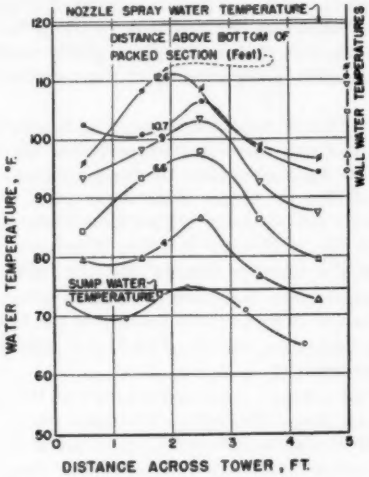


FIG. 5. TYPICAL TEMPERATURE DISTRIBUTION FOR COOLING TOWER

Temperature measured along center line.
L = 32,500 lb per hr
G = 63,000 lb per hr
Inlet air 60 F dry bulb, 57 F wet bulb.

$$\frac{L\,w_s\left(\frac{A_g - A_n}{A_g}\right)}{w_d \quad S}$$

does not have a singularity. The value of

$$\frac{L\left(\frac{w_s}{w_d}\right)\left(\frac{A_g - A_n}{A_g}\right)}{S}$$

becomes

$$(67000)\ (0.4)\ \frac{1.11\left(\frac{24.2 - 20}{24.2}\right)}{2880} = 17.9 \text{ lb/ (hour) (square foot)}$$

for the present experimentation at the point referred to previously on Fig. 3 where partial wetting begins. In the case of the experimental data presented by London, Mason and Boelter[1], the value of $\frac{L(w_s/w_d)[(A_g-A_n)/A_g]}{S}$ is 3.8 lb per (hour) (square foot) for the critical point of wetting they reported. The quantity w_s/w_d was assumed to be about 1.0 for the London, Mason and Boelter data. Comparison of the two values shows that the present method of determining the critical point yields a value five times the value obtained by the former group. It is believed that the step made in trying to correlate the critical points of the two towers, even with seemingly negative results at present, should lead toward a more complete understanding of the correlation of other desirable tower variables.

Water Temperature Distribution: Observing Fig. 5, which represents a typical set of data, it was found that the water temperatures near or at the center of the tower were 5 to 20 deg higher than those near the sides. Higher temperatures may be expected near the center if the water rates are more concentrated there and also if the air velocities are higher toward the sides. In experiments now being conducted, the wall water is being eliminated near the top of the tower as much as possible in order that its effect be removed.

The wall water temperatures decreased by only 40 percent of the total of the water temperature change in the packing because of the lack of stirring, splashing, and surface area during its descent. The first is equivalent to a high water side resistance to heat transfer.

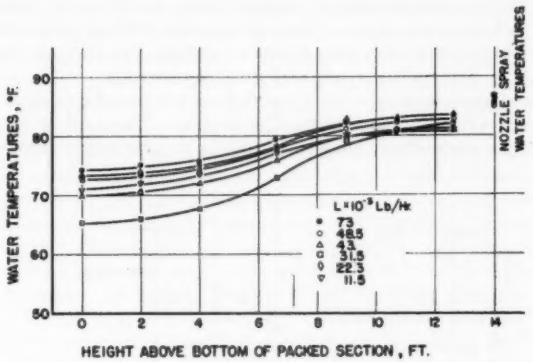
Fig. 5 shows that the average water temperature at the bottom of the packed section was lower than that of the sump. The sump temperature of 74.5 F was due to mixture of 95 F wall water and 70 F water from bottom of packing. Fig. 6 shows wall water temperatures as a function of height for Test Group 3.

Water Distribution: From the data which were obtained for the three Test Groups at different times, the percentage of wall water, slat water, and water drops between slats varied somewhat. The results were averaged and yielded the following data:

PERCENT WATER RATES

Test Group	Wall	Slats	Drops
1	40	25	35
2	41	31	28
3	28	32	40
Av'g. of 1, 2, & 3	36	30	34
Av'g. of 1 & 2	40	28	32

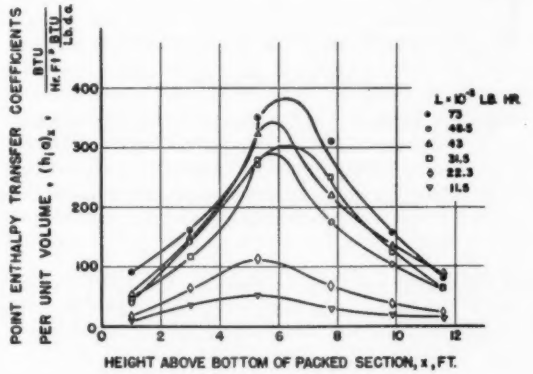
The slat and drop water data were obtained at one place in the tower with the wall water in a separate device but all results were corrected so that the sum of the three quantities equaled the total measured water rate. Since the average results given in the last two rows were similar, it may reasonably be concluded, for the present, that the water rate distribution was approximately divided in percent as follows: wall water = 40; packing slat film water = 30 and water drops = 30.



Air rate=76,800 (±11 percent) lb per hr.
Data from Test Group 3.

FIG. 6. WALL WATER TEMPERATURES AS A FUNCTION OF HEIGHT

Of the water in the packed section only, an equal division between drops and slat film occurred. It is not justified, therefore, to assume that the surface area of heat or mass transfer is only the packing surface area, as was done in the work presented by London, Mason and Boelter[1], because up to half the water in the tower at any particular time was not in the form of packing film water.

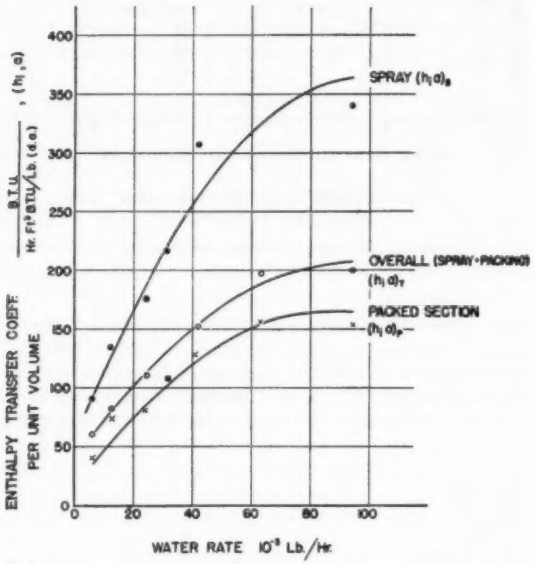


Air rate=76,800 (±11 percent) lb per hr.
Data from Test Group 3.

FIG. 7. POINT ENTHALPY TRANSFER COEFFICIENT PER UNIT VOLUME AS A FUNCTION OF HEIGHT IN PACKED SECTION FOR VARIOUS VALUES OF L

Transfer coefficients should then be reported as h_1a and not h_1 which is based only upon a packing surface area. Even though the ratio of water rate by drops to water rate as packing film may remain constant, the surface area of water drops to packing film will increase with total water rate.

Research and analysis is now being carried on to show the relative importance of water drop cooling to the packing film cooling. This will be illustrated not only by making a microscopic analysis on the individual water drop and film, but



Comparison between spray and packed section performance.
Air rate=67,000 (±19 percent) lb per hr.
Data from Test Group 2.

FIG. 8. OVERALL ENTHALPY TRANSFER COEFFICIENT PER UNIT VOLUME

also by varying the density and types of packing. The latter will serve to change the performance of the tower by causing different heights of fall of the water drops and also different drop size distributions due to splashing.

Point Enthalpy Transfer Coefficient per Unit Volume: Point enthalpy transfer coefficients per unit volume are shown plotted as a function of the position x which is measured above the bottom of the packed section at points halfway between the water temperature movable thermocouple stations. Data for Test Group 3 were used in drawing Fig. 7. Fig. 7 showed the tendency of point conductances to be highest approximately halfway up the packed section.

It has been noted in the heat transfer from tube walls to fluids flowing into

and through a tube that the point transfer coefficients were high at entrances because of the fluid flow conditions. Uniform velocity distributions at the entrance which change into turbulent or viscous velocity distributions further into the tube were found to be the cause. Water and air distributions in the cooling tower were far from being uniform. Air entered the tower by making a right angle turn and water entered from nine individual sprays. The same conditions as mentioned for heat transfer were not found in the cooling tower but it is believed that velocity distributions were the cause of the varying transfer coefficients shown in Fig. 7. It might be guessed that the point transfer coefficients should be higher at the water entrance side of the packing because the water temperatures were higher and air flow more uniform. But in observing the water sprays some time after tests were made, it was noted that water was not evenly distributed on the packing because of the sheet-like characteristics of the nozzle spray. It may be concluded that the water distribution at the spray end of the packing was uneven and therefore would not be conducive to high transfer coefficients. Lower point transfer coefficients at the ends of the packed section were probably caused by the non-uniform distribution of air and water at the respective entrances.

Further experimentation should attempt to verify these results and eliminate or correct for the end conditions mentioned. For the case of water distribution, a set of screens may be used just after the sprays in order that the water drops be spread more evenly. Air flow conditions at entrance should be measured before specific conclusions are made. If a non-uniform air flow exists, there would be great difficulty making corrections with the present tower. A long calming section would be needed to obtain uniform entrance conditions. Certain end conditions simulating commercial operating tower conditions are, however, desirable.

Overall Transfer Coefficient per Unit Volume: In order to make a comparison between the cooling due to the spray between the nozzles and packing and the cooling due to the packing, an overall unit volume transfer coefficient was calculated for each part respectively. Fig. 8 shows results for Test Group 2 for which overall unit volume conductances for the spray section, packed section and spray and packed section were plotted *vs* the water rate with the air rate being fairly constant for all runs.

Enthalpy transfer coefficients per unit spray section volume as shown in Fig. 8 were from 200 to 230 percent greater in magnitude than the unit volume transfer coefficients for the packed section only. Note the sharp increase in enthalpy of the air for this data as shown in Fig. 3 at the various water rates.

Drops falling in the spray section were traveling at high velocities and existed in a high state of agitation after breaking from the turbulent sheets of water near the spray nozzle tips[7]. Drops formed at the bottom of packing slats fell from a zero initial velocity and a relatively quiet state of agitation as compared to the spray drops. Two effects of heat transfer in falling water drops are noteworthy:[7,8]

1. Heat transfer from drops to air increases with the relative velocity of the drops to the air.

2. Liquid side resistance to heat transfer from drops reduces with increased convection or agitation within the drops.

With these considerations, conductances may be expected to be higher in the spray section and this was verified by the data plotted in Fig. 8.

An overall unit volume transfer coefficient was calculated for the whole tower, spray plus packing, and is also shown in Fig. 8. It may be seen that this curve is closer to the packed section curve which is explained by the fact that the volume of the packed section was approximately ten times that of the spray section.

Components of Packing Cooling Due to Drops, Slat Film and Wall Water: An attempt was made to obtain data in order that computation could be made for the difference in cooling in the packed section due to drops and that due to slat films. This was done by utilizing water temperature measuring devices which segregated the two forms of water. Film temperatures at the bottom of each slat were found to be up to 1 deg higher than the adjacent drops. Of the data taken, none were representative enough to correlate with the other results.

Pressure Drops: Only a small amount of pressure drop data was obtained along with the heat transfer data. Since the groups of test runs were made at essentially constant air rates and because pressure drops vary with the square of the air rates, no extensive correlation could be made.

With the data obtained in Test Group 2, it was found that pressure drops varied from 0.2 to 0.35 in. of water for a variation of water rate from 6500 to 95,500 lb per hour when the average air rate was 67,000 lb per hour. A friction factor, calculated from these results by the use of the Fanning equation,

$$\frac{f}{R}\frac{L}{}\frac{u^2}{2g} = \Delta p \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

where

f = friction factor.
L = length of packed section, feet.
R = hydraulic radius (open cross sectional area/perimeter of packing slats in one deck), feet.
u = mean air velocity through packing, feet per second.
Δp = pressure drop of air through packed section, feet of air.

varied from 0.40 to 0.70 respectively. The Reynolds number was approximately 50,000 based on an hydraulic radius of the net cross sectional area.

CONCLUSIONS

1. Air enthalpies decreased sharply with water rates at all positions in the packed section for value of L/G from 0.35 to 0.4, indicating a possible critical point for partially wetted packing.

2. The water temperatures over any cross section in the packed section were maximum approximately midway between the tower sides. Values were found to be from 5 to 20 deg higher than those near the sides.

3. The temperature of the water film on the walls was found to decrease by only 40 percent of the decrease in temperature of the packing water over the height of the packed section.

4. The total water flowing was found to be divided between the film on the walls, drops between packing slats and film on the packing slats in percent as follows: wall water film = 40; water drops between packing slats = 30; water film on packing = 30.

5. The point enthalpy transfer coefficients per unit volume, $(h_i a)_x$, were maximum approximately halfway between the ends of the packed section.

6. Overall enthalpy transfer coefficients per unit volume of the spray section above the packing were from 185 to 235 percent greater than for those in the packed section. Overall enthalpy transfer coefficients per unit volume were found to vary from 85 to 360 Btu per (hr) (cu ft) (Btu per lb dry air) for the sprays and 40 to 160 Btu per (hr) (cu ft) (Btu per lb dry air) for the packing over a water rate range of 6500 to 95,500 lb per hr. Air rates remained approximately 67,000 lb per hr $\pm$ 19 percent for all runs.

7. For the range of water flow of 6500 to 95,500 lb per hr and an average air rate of 67,000 lb per hr, the pressure drops varied from 0.2 to 0.35 in. of water respectively. Correspondingly, the computed Fanning friction factors were found to vary from 0.40 to 0.70. The Reynolds number for air was approximately 50,000 based upon an hydraulic radius of the net cross sectional area.

NOMENCLATURE

A_g = gross cross sectional area of tower, square feet.

A_n = net or open cross sectional area of packed section, square feet.

a = surface area of energy transfer per unit volume of tower, square feet per cubic foot.

C_{pw} = specific heat of water, Btu per (pound) (Fahrenheit degree).

h_i = unit enthalpy conductance, Btu per (hour) (square foot) (Btu per pound of dry air).

$(h_i a)_p$ = overall enthalpy transfer coefficient per unit volume of packed section, Btu per (hour) (cubic foot) (Btu per pound of dry air).

$(h_i a)_s$ = overall enthalpy transfer coefficient per unit volume of spray section, Btu per (hour) (cubic foot) (Btu per pound of dry air).

$(h_i a)_T$ = overall enthalpy transfer coefficient per unit volume of spray and packed section, Btu per (hour) (cubic foot) (Btu per pound of dry air).

$(h_i a)_x$ = point enthalpy transfer coefficient per unit volume at height x, Btu per (hour) (cubic foot) (Btu per pound of dry air).

g = gravitational acceleration, feet per (second) (second).

G = air rate, pounds per hour.

i_{ax} = enthalpy of air in tower at height x, Btu per pound dry air.

$(\Delta i_x)_{lm}$ = log mean enthalpy potential difference between two successive temperature stations in tower, Btu per pound of dry air.

Δp = pressure drop of air through packed section, feet of air.

L = water rate, pounds per hour.

R = hydraulic radius = $A_n \div$ perimeter (feet) of packing slats in one deck.

S = total surface area of packing in tower, square feet.

$(\Delta t)_x$ = temperature increment between two successive heights in tower, Fahrenheit degrees.

u = mean air velocity through packing, feet per second.

V_x = volume of tower included between two successive heights in tower, cubic feet.

w_s = water rate as film on packing slats, pounds per hour.

w_d = water flow between slats in the form of drops, pounds per hour.

x = height above bottom of packed section, feet.

f = friction factor defined by the Fanning equation.

ACKNOWLEDGMENT

Plans for the tower were originally made by L. M. K. Boelter, E. D. Howe, and D. D. Davis. Messrs. C. Miller, H. Eagles, and J. DeCosta assisted in construction of the tower. Greatly appreciated were the experimental assistance of M. Greenfield, B. G. Case, C. H. Bailey, H. Ast, D. Billings, B. Mahin, and F. McInerney. Mr. Greenfield was of great assistance in the compilation and computation of the data. Guidance was extended by L. M. K. Boelter and E. D. Howe.

The cooperation of the *Cooling Tower Institute,* the assistance of the Redwood Manufacturers Co., the Marley Co., Inc., and W. R. Ames Co. in supplying equipment, and the active interest of E. Simons and L. T. Mart in obtaining equipment for the tests, are acknowledged with appreciation.

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DISCUSSION

H. E. DEGLER, Kansas City, Kans. (WRITTEN): This experimental tower has nine 1½ in. spray nozzles with about 20-in. spacing and downspray water distribution, whereas wood-filled *conventional* counterflow designs generally use ¾ to 1¼-in. *up-spray* nozzles set at 15 to 24-in. centers and nozzle pressures of 3 to 4 psi to insure good water breakup. By calculation, the tower has about 10 sq ft of wood-fill surface per cubic foot of active tower volume, whereas actual towers have wood-fill concentrations of less than half this amount. A plan area of at least 6 x 6 ft would have more nearly conformed with practice. By reducing the height and increasing the plan area, the same amount of fill could have been advantageously expanded to a more generally used concentration of 3 to 5 sq ft of surface per cubic foot of tower volume, and with a reduced draft loss.

The maximum ratio of $\frac{L}{G} = 1.6$ (Fig. 3) is comparable with practice. Higher liquid-gas ratios indicate little change in performance, which is predictable from re-

sults at lower ratios. Experimental work further indicates that liquid-gas ratios lower than 0.4 can be used with some designs, probably as low as 0.25, with about the same tower performance. From Fig. 5, it is evident that a high water concentration existed down the center of the tower as well as along the walls, or perhaps a high velocity air vortex was moving up through the tower. The best cooling area was about ¾ of the distance from the center to the walls, where there existed a lower (more desirable) $\frac{L}{G}$ — ratio. The point unit-volume enthalpy transfer coefficient would probably have been higher nearer the top of the tower (Fig. 7) and undoubtedly a higher overall value obtained (because of better drop dispersion) if an up-spray system had been used. It would be of interest to know what nozzle pressure was used when operating the tower at the capacity of 95,500 lb of water per hour (191 gpm).

The work shows good instrumentation, but the tower does not simulate actual design because of (1) its dimensions, (2) large amount of woodfill, (3) use of downsprays, and (4) intensified water concentration along the center and at the walls (as indicated by the highest temperatures, center and sides, in Fig. 5). It would be interesting to know the effect of streamlined egg-crate (channeled and stratified) air-stream flow immediately after the air enters the bottom of the tower. This experimental water-cooling tower could be quite adaptable for a subsequent investigation of the effect of changing the wet bulb temperature of the air entering the tower. Some of the moist air discharged from the fan could be readily conducted to the base of the tower in such amounts as to obtain any desired wet bulb variations of the inlet air. This is basic research that would be of value to all persons interested in the cooling of water with atmospheric air.

F. EDWARD INCE, St. Louis, Mo.: With reference to Mr. Degler's comment, I feel that Professor Snyder's objective was the performance of a packed tower and not that of a commercially arranged tower. Surface performance through this extreme height was not what could be expected from a commercial unit.

The packing is of little consequence in the total capacity. This is apparent by the performance shown by the spray nozzle in its enthalpy capacity at that point. Perhaps a series of nozzles will be better than having any packing at all. If packing is proven useful, perhaps we should eliminate the nozzles entirely in future programs and see exactly what becomes of this arrangement.

AUTHOR'S CLOSURE: In answer to Mr. Degler, the fact that the density of packing in this tower is twice that of some commercial towers does not necessarily mean that there was no justification for the tests. In fact, an extension was made in the density in order to find the effect upon the performance. Subsequently it is planned to present the performance as a function of packing density based on work already completed. In the recent paper by Simpson and Sherwood (see Reference 6) even greater densities of slat packing were reported and shown to be of considerable advantage both with respect to performance and pressure drop.

There is no particular magic in using a tower 6 x 6 ft for these tests because it happens to be one of the convenient commercial sizes. Many small commercial towers have inside dimensions other than an exact 6 x 6 ft, including towers from Mr. Degler's company.

The variation in water temperatures across the tower was undoubtedly due to an adverse water distribution which, in turn, influenced the air distribution. An upspray system would help in causing a more even water distribution. However, my experiments show (Fig 8) that a downspray within 15 in. of the nozzles exhibits very high performances. I have more data for sprays which I shall publish at a later date, showing enthalpy transfer coefficients per unit volume as high as 600 Btu per (hr)

(cu ft) (Btu per lb dry air). Therefore, it would be important to know the effect of the downsprays upon the packing performance. A more judicious choice of spray-nozzle sizes and placement would yield an optimum performance in a downspray-packed tower.

I am carrying out a study on the effect of the air-entrance construction upon the flow of air into the tower.

Changing the wet bulb of the entering air would have little or no effect upon the enthalpy transfer coefficient per unit volume. The transfer coefficient varies particularly with the air rate and water rate and is somewhat affected by the entering water temperature.

In answer to Mr. Ince, as I have pointed out and from the results in the paper concerning the performance at the spray nozzles, it would be advantageous to build a small tower with sprays. Dispersing these spray nozzles with height in the tower may not be the best arrangement as compared to placing the spray nozzles at one cross section. As the air gains enthalpy in passing through the tower, the cooling range of the water from each subsequent set of spray nozzles decreases. This problem requires further study.

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No. 1353

HEAT TRANSFER RATES FOR REFRIGERANT BOILING IN HORIZONTAL TUBE EVAPORATORS

By L. G. SEIGEL*, W. L. BRYAN**, AND M. C. HUPPERT†, CLEVELAND, OHIO

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the *Case Institute of Technology*, Cleveland, Ohio

IN SPITE of the general use of dichlorodifluoromethane (Freon-12) as a refrigerant in direct expansion heat exchangers, little basic information is at hand regarding its heat transfer behavior. Generally, it has been assumed that evaporating Freon has some average heat transfer coefficient and then this coefficient has been applied to heat transfer equipment to obtain performance data. This has sometimes resulted in erratic performance data and in some cases may actually lead to the selection of improper equipment for a given service. Therefore, the object of this investigation was to examine and to correlate the factors which seem to have the most significant effect on heat transfer such as loading, oil circulation with the Freon, L/D ratio, refrigerant flow, and temperature difference, tube to refrigerant.

SCOPE OF INVESTIGATION

This paper gives the results of a series of tests on three simple straight tube evaporators. It was decided to begin the investigation with these tests in view of the fact that industrial evaporators often are complicated by methods of circulating and of loading. Therefore, the data presented here represent essentially basic performance for evaporating Freon-12 in single tubes. Application to complicated heat exchangers should be made with due caution. In these tests, no attempt was made to control the amount of oil circulating with the Freon. However, tests were run both with and without an oil separator and attempts were made to measure the oil quantity in circulation. Tests were also run to determine the effect of excess liquid at the evaporator outlet. In all cases, subcooled liquid was supplied to the expansion valve. Saturation temperatures in the evaporators ranged from 30 to 80 F.

DESCRIPTION OF TEST EQUIPMENT

The equipment used for the purpose of these tests consisted of a one ton reciprocating type Freon condensing unit equipped with a variable speed drive

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† Graduate Student, Case Institute of Technology.

Presented at the 55th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1949.

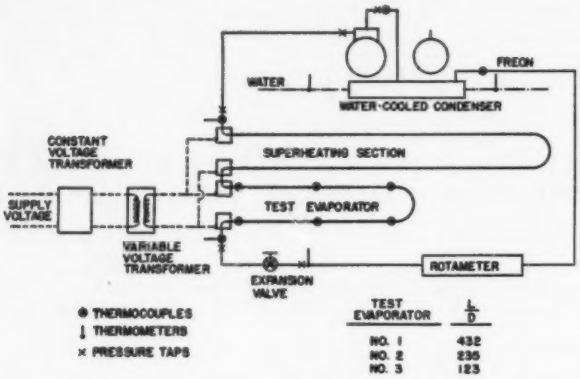

FIG. 1. SCHEMATIC DIAGRAM OF FREON-12 HEAT TRANSFER TEST SETUP

and a water cooled condenser. The condenser was insulated so that a heat balance could be obtained and used as a check on the heat transfer calculation for the evaporator. A schematic diagram of the set-up and arrangement of evaporators is shown in Fig. 1.

Freon and water temperature measurements were made with matched thermometers capable of being read to 0.1 F deg (0.2 deg graduations), and tube temperatures were measured by means of calibrated copper-constantan thermocouples attached directly to the tubes according to good practice[1]. The entire thermocouple system was calibrated before use by comparison with the same matched thermometers used in the Freon circuit. Also as a running check on the thermocouple system, check couples were installed at the same locations as certain mercury thermometers. Evaporator pressure measurements were taken through a manifold system connected to a common gage so that a reliable pressure difference would be obtained.

All evaporators were made of $\frac{5}{8}$ in. O.D. copper tube. Those referred to as Nos. 1, 2 and 3 had one return bend and had L/D ratios of 432, 235 and 123, respectively. One water heated evaporator for which results are also shown consisted of a $\frac{5}{8}$ in. O.D. straight copper tube.

METHOD AND PROCEDURE

Each test run consisted of a set of 5-min readings of all instruments for a period of approximately one hour after equilibrium conditions had been obtained. Evaporator suction temperature was taken as the saturation temperature corresponding to the suction pressure. However, this method of obtaining evaporator temperature was under constant check by thermocouple, and was accepted only after preliminary tests indicated that the charge of Freon in the

[1] Temperature, Its Measurement and Control in Science and Industry—Sponsored by Symposium Committee of *American Institute of Physics*. (Reinhold Publishing Corp., N. Y., 1941.)

system actually agreed with the published data for Freon-12. A heat balance on the water cooled condenser was computed for each run and the flow of Freon was obtained from this balance. This Freon flow was then applied to the enthalpy rise across the evaporator to obtain the evaporator load. Plots of transformer current against evaporator load were made for each evaporator for all runs in which the Freon was superheated leaving the evaporator. This provided curves of current flow vs. load which was used to determine evaporator loads for tests in which the Freon left the evaporator as a wet mixture.

A rotameter installed in the liquid line before the expansion valve was also calibrated during the tests which were run with superheated vapor leaving the evaporator. This calibration permitted the evaluation of the Freon flow for tests run under conditions of a wet mixture.

The rotameter tube also provided a means for determining the specific gravity of the liquid before the expansion valve by inserting a hydrometer float in

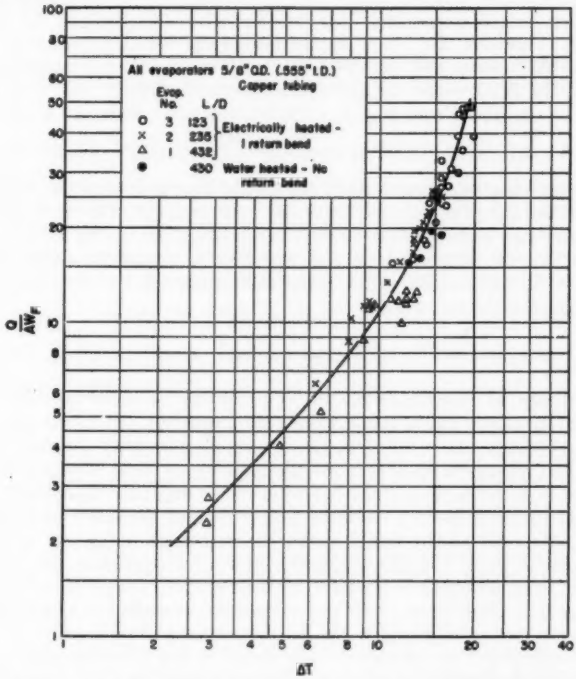

FIG. 2. HEAT TRANSFER RATES FOR FREON-12 IN FOUR HORIZONTAL TUBULAR EVAPORATORS

$\frac{Q}{AW_F}$ = Heat transfer Btu per (hour) (square foot internal surface area) (pound Freon-12 flow per hour)

Δt = Temperature difference between tube and refrigerant, Fahrenheit degrees

place of the rotameter float. This was done in an effort to observe any change in specific gravity which would indicate the presence of oil. However, no appreciable variation in specific gravity was observed.

Condenser water temperature measurements were made with thermometers which had 0.2 deg graduations and had a calibration accuracy of ±0.1 F deg. All other temperature measurements were made with copper-constantan thermocouples connected to an electronic potentiometer. The scale of this instrument could be read to ±0.1 F deg. The thermocouples were calibrated in place with thermometers of ±0.1 F deg accuracy.

RESULTS

The experimental results for the three electrically heated evaporators are shown in Fig. 2. Also shown on the same figure for comparison are four test points from a water heated evaporator previously tested. The tube diameter of the water heated evaporator was the same as was used in the electrical evaporators. However, the water heated evaporator was a straight ⅝ in. O.D. copper tube 20 ft long, and therefore, the L/D value did not include a return bend as did the L/D values for the electrical evaporators.

As may be observed from Fig. 2, a correlation was obtained for all evaporators when the data were plotted in terms of $\frac{Q}{AW_F}$ *vs.* Δt^*. All the data plotted in Fig. 2 were obtained from tests in which the Freon left the test section with a quality in the range of approximately 30 to 80 percent. However, the results of other tests run with approximately 10 deg superheat at the evaporator outlet indicate that the correlation of Fig. 2 is reliable within the limits of experimental error throughout the entire evaporation range up to 100 percent quality. These results are shown in Table 1.

EFFECT OF OIL

In an attempt to investigate the effect of oil on the heat transfer rate, the specific gravity of the Freon liquid before the expansion valve was measured by means of a hydrometer installed in the rotameter tube after the float had been removed. No variation in either specific gravity or heat transfer rate could be observed in a series of tests during which the compressor speed was changed in an effort to carry over a variable quantity of oil for each test. However, changes in the color of the liquid before the expansion valve and the readings of thermometers located at the evaporator outlet indicated that the oil quantity did actually vary, although it produced no appreciable change in specific gravity or heat transfer. Therefore, an oil separator was installed in the system after the tests were run on evaporator No. 1. Evaporators Nos. 2 and 3 were tested with the oil separator in operation. The liquid before the expansion valve was clear in all tests after the installation of the oil separator and no further difficulty was encountered with the thermometer readings. These facts indicate that less oil was in circulation during the tests of evaporators Nos. 2 and 3 than during the earlier tests of evaporator No. 1. However, there was still no noticeable change in heat transfer rate as indicated by the relatively good agreement of all results shown in Fig. 2. The somewhat greater scatter of the points

* See list of symbols.

TABLE 1. COMPARISON OF RESULTS FOR SUPERHEATED VAPOR WITH THOSE FOR WET MIXTURE[a]

EVAPORATOR TEMPERATURE F	$\frac{Q}{A}$	$\frac{Q}{AW_F}$	FULL TUBE LENGTH		HALF TUBE LENGTH	
			Δt	Superheat F Deg	Δt	Superheat F Deg
30.1	1945	19.7	20.2	11.9	19.1	WET MIXTURE
39.3	1900	19.9	16.6	8.7	15.7	
49.6	1875	20.0	17.3	9.7	15.8	
56.7	1810	20.4	19.0	9.1	15.7	
30.6	1950	19.8	17.0	10.0	16.8	
41.5	1900	20.1	20.6	14.0	16.0	
48.9	1915	20.3	17.6	10.5	15.8	
56.1	1820	20.7	18.3	10.8	16.7	
46.0	1890	20.0	17.4	9.3	15.6	
18.3	1760	19.2	13.9	10.6	14.8	

[a] From tests of Evaporator No. 1 before addition of superheater section.

for evaporator No. 1 may have been caused by inaccurate temperatures. It was found to be difficult to obtain accurate temperature measurements with oil present.

EFFECTS OF FLASH GAS AND QUALITY

Flash gas entering the evaporators ranged from 8.5 to 20 percent and no appreciable effect was noticed due to its presence when the data were correlated as shown in Fig. 2. Approximately the same amount of *scatter* of points resulted in data plotted at constant flash gas as in those plotted without consideration of flash gas. However, it should be recognized that the ordinate of Fig. 2 actually accounts for variable flash gas since $\frac{Q}{W_F}$ is really the heat absorbed per pound of Freon. In cases where the Freon leaves the evaporator at 100 percent quality $\frac{Q}{W_F}$ varies with the flash gas percentage for a given pressure.

It is interesting to examine further the ordinate of Fig. 2 for conditions of 100 percent quality leaving the evaporator and 0 percent flash gas entering the evaporator. Under these conditions, $\frac{Q}{W_F}$ becomes equal to the latent heat of vaporization and the ordinate for a given evaporator pressure and area becomes a maximum. For Freon-12 the heat of vaporization is nearly constant in the range of normal evaporator operation. Therefore, $\frac{Q}{AW_F}$ as well as Δt, must also be nearly constant for a given evaporator. Fig. 3 is an attempt to demonstrate this fact. The scatter of the points along the line of constant Δt (line A-B) in this figure may be attributed to an inability to maintain 100 percent quality and to obtain 0 percent flash gas.

CONCLUSIONS AND RECOMMENDATIONS

1. The heat transfer rate for Freon-12 evaporating in horizontal tubes may be correlated in terms of Q/AW_F *vs.* Δt.

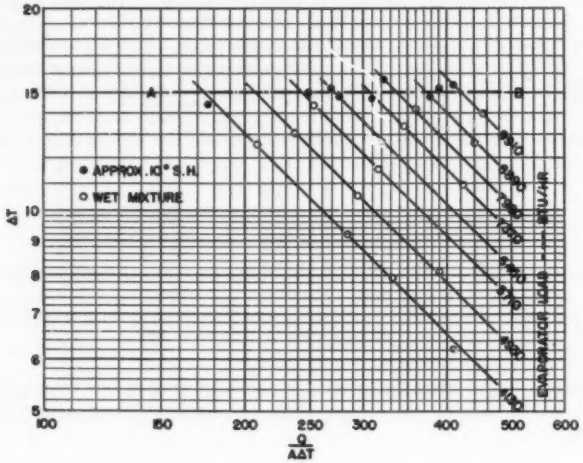

FIG. 3. VARIATION IN HEAT TRANSFER COEFFICIENT WITH Δt AND EVAPORATOR LOAD (FROM TESTS OF EVAPORATOR NO. 3)

Δt = Temperature difference between tube and refrigerant, Fahrenheit degrees

$\frac{Q}{A\Delta t}$ = Heat transfer, Btu per (hour) (square foot of internal surface area) (Fahrenheit degree difference between tube and refrigerant)

Additional tests should be run with different tube diameters.

2. The presence of oil in Freon-12 evaporators does not seem detrimental to heat transfer when the oil is present in quantities such as obtained with ordinary commercial equipment. However, quantitative tests should be run to determine how much oil can be tolerated before the heat transfer is affected.

3. L/D ratios from 123 to 432 had no effect on heat transfer. However, only one tube diameter was used in this investigation and, therefore, similar data should be obtained for other tube diameters.

4. Shorter tube lengths (smaller evaporator areas) gave greater values of $\frac{Q}{A}$ and Δt. Still smaller evaporators should be investigated to determine limiting conditions.

LIST OF SYMBOLS

A = Internal surface area of evaporator, square feet

Q = Total heat transferred, Btu per hour

W_F = Freon-12 flow, pounds per hour

Δt = Temperature difference between tube and refrigerant, Fahrenheit degrees

L/D = Ratio of heated length to inside diameter

DISCUSSION

H. B. NOTTAGE, Cleveland, Ohio (WRITTEN): If this paper is to be published in the A.S.H.V.E. TRANSACTIONS, complete data should be added for each run. These should include: (1) The initial and final state of the refrigerant; (2) refrigerant flow rate; (3) heat transfer rate; and (4) tube inner surface-temperature distribution.

How was the tube temperature for Fig. 3 defined and measured experimentally?

More details on the heat-loss calibration would be of interest. Did this loss vary in simple relationship to the rate of heat generation, evaporator load, and the temperature difference to the surroundings?

Use of electrical tube heating would mean a uniform heat load per unit of tube length except for variations in loss to the surroundings, longitudinal conduction, and the distortion at the bend. The lengthwise distribution of tube temperature should then allow an estimate of the lengthwise variation of the unit conductance for tube-to-refrigerant heat transfer. Such magnitudes would be helpful in design and also give better evidence on the influence of quality upon the heat-transfer rate. An effect of quality would be expected because of the change in fluid properties accompanying vaporization, and it may be that this was masked by the large L/D ratios employed.

Fig. 2 shows that the unit enthalpy increase for the evaporating freon depends primarily upon the tube surface area and Δt. Should not this be more explicitly limited to the test range of W_F?

The straight lines on Fig. 3 are, in effect, plots of Δt vs. $1/\Delta t$. Scatter of test points would indicate non-constancy of load. If the electrical input was held constant for each line, then experimental evidence exists that the portion of the input lost to the surroundings was slightly influenced by conditions in the refrigerant passage.

C. M. ASHLEY, Syracuse, N. Y.: This is a significant paper as it measurably increases knowledge in an area which has been badly neglected in the past. Presenting, as it does, the data for several lengths of tubes and over a range of refrigerant quality, it permits a direct correlation of variables other than those which were possible with my test data which were presented in 1941. These were based upon tests of a single tube length and largely at saturated leaving conditions.

It is of special interest to compare the results of these two studies. The tests reported by me show temperature differences of approximately 30 percent less than those shown by Fig. 2 of the present paper for refrigerant leaving the evaporator saturated; and approximately 10 percent less for refrigerant leaving the evaporator with 70 to 75 percent quality. However, the evaporator length for my tests was 60 percent longer in terms of $\frac{L}{D}$ ratio than the longest evaporator tested in the present series of tests, and was 4 percent larger in diameter. It is of some interest to note that, when my data are corrected back to the length of the longest evaporator of the present test by the use of the tentative correlation suggested in the earlier paper, the value of Δt falls on the line of Fig. 2 for the saturated leaving conditions. Thus, for shorter evaporators, the use of the data from the earlier paper would be at least as conservative as that of the present paper.

Another point of interest is that apparently most of the data shown in the present test were taken at relatively low loads; that is, below 10,000 Btu per hr. In this region, my test data indicated a fixed value of Δt for a given evaporator at saturated leaving conditions which confirms the present series of tests. However, at higher values of loading, from 12,000 to 18,000 Btu per hour, the value of Δt is shown by the earlier test to decrease with increased loading. On this basis, I believe that the validity of the present series of tests must be considered as limited to low loading. They could, however, be used conservatively for higher loading.

I cannot account for the differences in results between my tests and those of the present paper. The difference in the length and diameter of the tube, however, may

be of some significance. The deviation from the curve of Fig. 2 is less than the total spread of the points. It is hoped that the authors will extend their work to cover longer circuits and higher loading rates. I recommend that they include in their final paper data on Δt at the different points through the test evaporator for typical tests. These data will be of great value as a cross-check and as a cross-interpretation of the system of correlation used.

The authors apparently used no refrigerant pressure and temperature measuring stations at the end of the evaporator. It would be of interest to know what the total pressure drop through the evaporator and superheating sections were. To be of the greatest value, the data obviously should be based upon the mean refrigerant temperature, rather than that at either end of the evaporator. Information on the closeness of the check of evaporator load by the two methods used would also be of interest.

It is requested that essential original test data be included in the final draft of the paper. I note a growing tendency to omit these data for brevity and readability. This is to be deplored, because it makes the interpretation of data by the reader difficult, if not impossible. I have many times found it extremely valuable to make such an interpretation. The essential quality of the data is impaired by such omission.

F. EDWARD INCE, St. Louis., Mo.: In Fig. 3, are superheat or saturated temperatures involved for values of Δt in the range of 15?

AUTHORS' CLOSURE: The authors wish to thank the discussers for bringing out a number of interesting points. Several have mentioned inclusion of data. We would like to have included data, but it was a question of precedence that should be decided by the Society. The data alone would have filled the space given to the present paper. Should the paper be twice as long, including the data, or should section of the data be deleted? Our answer to this question was to include the results in curve form, increase the analysis and leave out the data. We felt that cutting down and putting in certain specific data would lead to wrong interpretations.

As to the six tube temperatures, they were measured by calibrated copper-constantan thermocouples attached directly to the outside surface of the tube in the positions shown in Fig. 1.

We agree with Mr. Nottage's comment on Fig. 3, that the scatter of points indicate fluctuations in load. These data (tests of evaporator No. 3) were taken prior to the installation of a constant voltage transformer so that constant electrical input was only approximated. It will be noted, however, that these data correlate with the other data as indicated by Fig. 2. It should be pointed out that the purpose of Fig. 3, was to explain the coordinates of Fig. 2 and not a presentation of data.

We appreciate Mr. Ashley's comments and the thought which he gave to our paper. Concerning the higher loads, we were limited with our size tubes to 12,000 Btu per hr load. However, on the basis of Btu per hr, sq ft, we obtained 9900. If calculated correctly, Mr. Ashley's range was 3700 to 12,000 Btu per hr, sq ft. Mr. Ashley's highest load in Btu per hr was 18,000.

It will be necessary, of course, to increase the length of the tube to explore what takes place beyond the 9000 Btu per hr load.

The pressure drop measured across the total evaporator was found to be very small and not of a sufficient magnitude to be measured accurately on a Bourdon type test gage. It was also found of insufficient magnitude to obtain consistent readings with a differential gage. Because of this, no attempt was made to measure the pressure drop across the test section of the evaporator.

Mr. Ince mentioned Fig. 3. The maximum temperature difference, tube-wall temperature minus saturation temperature of the refrigerant at evaporator pressure, was 15.3 F.

1354

No. 1354

PSYCHROMETRIC FACTORS IN THE AIR CONDITIONING ESTIMATE

By C. M. Ashley*, Syracuse, N. Y.

CURRENT methods of estimating summer air conditioning load and selecting equipment have many elements in common, but differ in directness of approach and assurance of results. The present paper describes one method[1] used by the author for fifteen years, and compares it with other methods. It is hoped that such comparison will stimulate discussion which may establish a uniform method of estimating and of equipment selection.

A sample estimate form which presents an overall view of the method is included as Fig. 1.

Approach to the Estimate

The estimator of a heat load for summer air conditioning customarily starts with certain *design conditions*. These conditions are usually those for which system performance is to be guaranteed and include:

a. Room air temperature and humidity.

b. Outside air temperature and humidity.

c. Number of people customarily in the space.

d. Internal sources of heat and moisture, such as lights, motors, cooking appliances, etc.

e. Ventilation standards to be maintained (usually in terms of cfm outside air per person).

Information must also be available as to the size of the room and the building construction. From the foregoing information the estimator determines (1) what heat and moisture load must be removed from the space to be con-

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[1] Exponent numerals refer to References.

Presented at the 55th Annual Meeting of The American Society of Heating and Ventilating Engineers, Chicago, Ill., January 1949.

ditioned; (2) what additional heat and moisture must be extracted from the ventilation air. Usually, the estimator also knows in advance the general type of air conditioning equipment to be used.

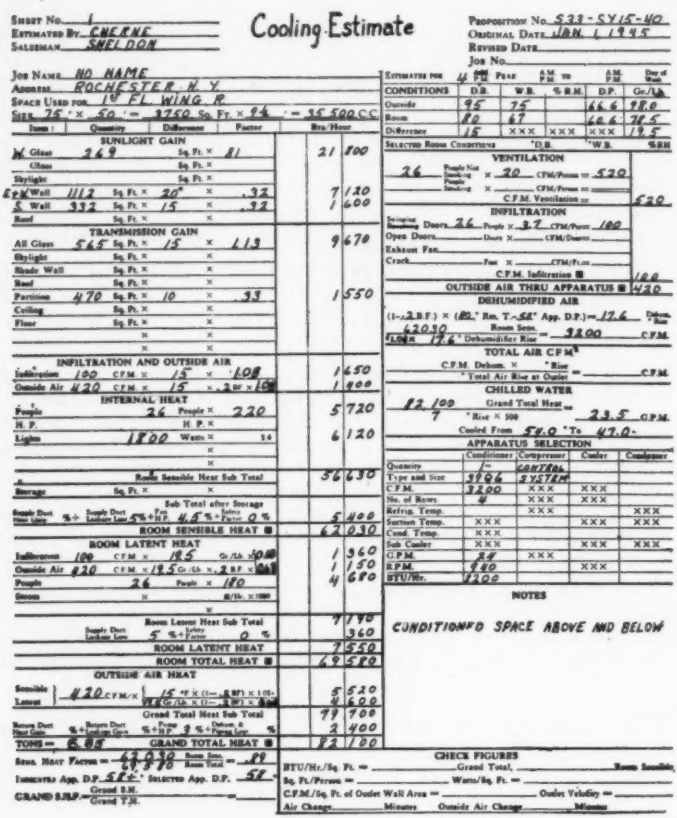

FIG. 1. COOLING ESTIMATE FORM

SENSIBLE AND LATENT HEAT

After determining the heat and moisture load of the room, the estimator next has to choose the temperature and humidity of the air to be supplied by the apparatus. He must also select the corresponding air quantity. The problem is illustrated graphically on the psychrometric chart (Fig. 2), where B represents the condition leaving the apparatus, and C the room condition. To reach

room conditions, the air, as it passes from the apparatus to the room, must absorb the heat and moisture load liberated in the room. This is illustrated by line BC.

The first part of the heat gain of the air is associated with a change of temperature and is generally termed *sensible heat*. The sensible heat gain is a function of: (1) the temperature rise of the air moving from the apparatus

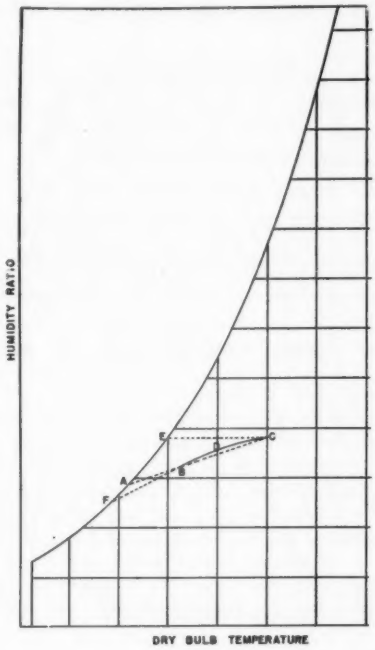

FIG. 2. PLOT OF AIR CONDITIONS ON PSYCHROMETRIC CHART

to the room; (2) the rate of air flow; (3) the specific heat of the air (including its moisture).

The second part of the heat gain is associated with a change of moisture content of the air (humidity ratio) and is commonly known as *latent heat*. The latent heat gain is a function of: (1) the increase in moisture content of the air as it goes from apparatus to room conditions; (2) the rate of air flow; (3) the *latent heat* of the moisture per pound. The sum of the sensible heat and latent heat is commonly known as *total heat*. In Appendix I it is shown that for the range of conditions used in comfort air conditioning, it is possible to consider as constants both the *specific heat* of moist air and the *latent heat* per pound of dry air. The error in total heat should be under 1 percent. This

is smaller than normally obtained by the use of uncorrected enthalpy at saturation.

SENSIBLE HEAT FACTOR

Since the same air is used for absorbing both moisture and *sensible* heat, the ratio of moisture to sensible heat gain per pound of dry air is the same as the ratio of the moisture load to sensible heat load to be absorbed in the room. The moisture content being the ordinate of Fig. 2 and the temperature the

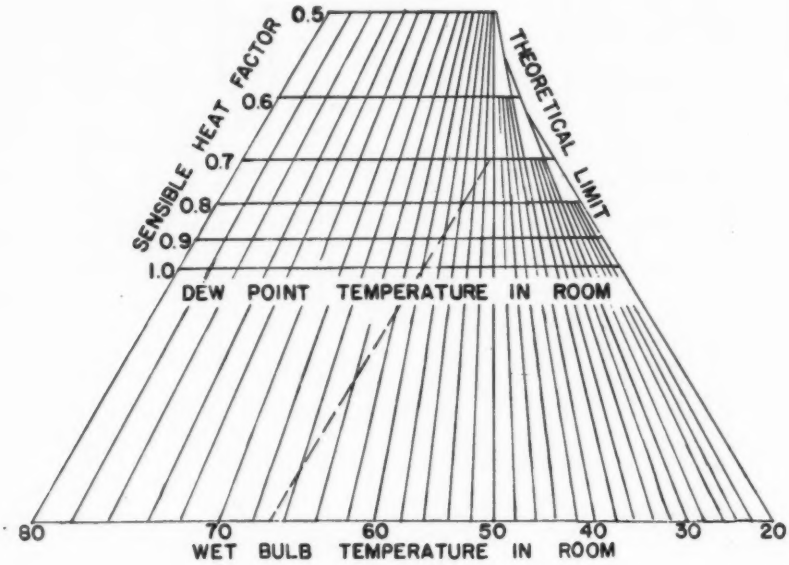

FIG. 3. APPARATUS DEW POINT CHART

abscissa, their ratio is expressed by a constant slope, ABC. This ratio can be expressed either directly or in terms of related functions. A convenient related function is the ratio of sensible heat to total heat. This is called the Sensible Heat Factor.

$$(\text{SHF}) = \frac{Q_s}{Q_T} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where:

Q_s = sensible heat load
Q_T = total heat load

On the psychrometric chart, the Sensible Heat Factor can be used to determine the slope of the line which joins the required state points of the supply

air leaving the apparatus and that of the room air. (See Appendix I, Equation 12*a*). Since this line represents the locus of points satisfying the given ratio of moisture to sensible heat load, the condition of the air leaving the apparatus can be chosen anywhere along this line. Thus, a means is obtained for determining the supply air conditions required to give the ***design*** room conditions. Any leaving apparatus conditions not on this line cannot satisfy the design room conditions.

APPARATUS DEW POINT

In Fig. 2, the *apparatus dew point* is at point A, where the sloping Sensible Heat Factor line corresponding to Sensible Heat Factor joins the saturation line. It should be understood that the supply air condition need not be at A, but can be at any point (such as B) on the line AC.

APPARATUS DEW POINT CHART

Since the slope of a line on the psychrometric chart (Fig. 2) is difficult to determine accurately, an apparatus dew point chart of the alignment type has been devised (Fig. 3). This chart makes possible the reading of the *apparatus dew point* within ± 1 deg.

COIL BYPASS FACTOR

In spray type dehumidifiers, the air leaving the dehumidifier is usually saturated and therefore is at the apparatus dew point. This simplifies the problem of choosing the correct apparatus dew point and air quantity.

When a coil is used as a dehumidifier, the air normally leaves in a condition less than saturated. This complicates matters, since the estimator must evaluate both the dry bulb temperature and the moisture content of the air leaving the coil. Moreover, these values are influenced by the condition of the air entering the coil (which may not be at the room condition).

Referring to Fig. 2, if C represents the entering state point of the cooling coil and B the leaving state point, then the coil apparatus dew point is found by extending the straight line joining points C and B to the saturation line at A. It should be understood that this line is not necessarily a *condition line* for the air flow through the coil (Fig. 4). Rather, it represents the relationship between the entering and leaving terminal conditions. The slope of this line gives the Sensible Heat Factor for the coil.

For a very deep coil with entering and leaving state points on this line, the leaving condition would be practically at saturation. The *efficiency* of the coil could be considered as 100 percent, and the *inefficiency* zero percent. With a shallower coil, this no longer holds true, and the final state point will be at some point such as B of Fig. 2. In this case, it might be assumed that a portion of the air came in contact with the coil surface and was reduced to the condition of the coil apparatus dew point, while another portion passed through the coil without any change of condition[2]. The portion assumed to have passed unchanged through the coil is termed the *bypass* portion. It might be considered as the *inefficiency* of the coil. The remainder is termed the *dehumidified* portion.

The Bypass Factor (Appendix II, Equation 16) can be described as the ratio:

$$\frac{\text{leaving air temperature minus coil apparatus dew point}}{\text{entering air temperature minus coil apparatus dew point}}$$

Since the conditions of the air entering the coil, leaving the coil, and at the coil apparatus dew point, are on a straight line on the psychrometric chart, the

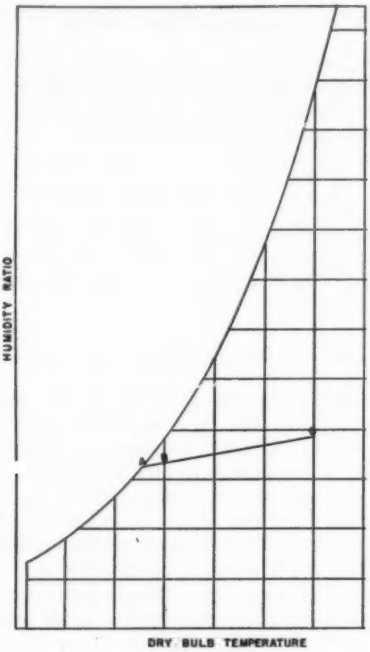

FIG. 4. CONDITION OF AIR PASSING THROUGH THE COIL

Bypass Factor with respect to temperature and moiture will have the same value.

$$\text{Bypass Factor (BF)} = \frac{t_2 - t_s}{t_1 - t_s} = \frac{W_2 - W_s}{W_1 - W_s} \quad \ldots\ldots\ldots \quad (2)$$

where: t_1, t_2 and t_s are the entering, leaving and apparatus dew point temperatures and W_1, W_2 and W_s are the corresponding humidity ratios.

COIL PERFORMANCE, AIR SIDE

The concepts of the coil Bypass Factor and of the coil *apparatus dew point* have been developed without any relation to the surface temperatures actually

existing in the coil. Such a relationship is desirable in solving problems, and was implied in Carrier's 1911 statement of the laws of psychrometry[3]. Later, he[4] and others[5, 6] developed the theory that, in any process involving simultaneous transfer of heat and of moisture between air and a wet surface, the transfer of heat and of moisture between the surface and air is in proportion to the respective heads.

This relationship is presumably due to the diffusion of the water vapor molecules into the air at substantially the same rate as the transfer of heat (kinetic) energy between the molecules. (Actually, the water vapor diffusion is somewhat faster.) Since the rate of transfer of heat and moisture can be considered the same with respect to their heads, the condition of transfer is the same as if a portion of the air were brought in contact with the surface, and its temperature and moisture content were brought to the surface condition and then remixed with air passing unchanged through the coil.

But this is exactly the analogy used previously to illustrate the relationship between the entering and leaving coil conditions and the coil apparatus dew point. It can then be seen that the coil apparatus dew point has a physical as well as a mathematical significance. It is the uniform coil surface temperature necessary to produce the actual combination of entering and leaving conditions (as shown by line AC, Fig. 2).

In practice, the coil surface temperature is seldom uniform throughout, particularly where the coil has extended surface on the air side. A typical condition of the air as it passes through the coil is represented by the curved line CDB, Fig. 4, with initial surface temperature at E and the final surface temperature at F. However, the leaving air condition B is the same as though the coil surface temperature were constant at A, (See Appendix II).

It is thus possible to assign a single surface temperature as the composite of actual temperatures. Since this *effective* surface temperature is the lower terminus of the sloping line on which the entering and leaving coil conditions are located, it can be considered as the coil *apparatus dew point.*

Fig. 5 shows typical Bypass Factors for a line of finned coils. (The derivation of the coil Bypass Factor is shown in Appendix II.) As will be seen, the Bypass Factor of a coil is a function primarily of the depth of the coil and, to a lesser extent, of the air velocity. It can be considered as a measure of the effective coil depth. The Bypass Factor thus indicates the air side performance of a chosen coil, and makes possible the calculation of the conditioned air quantity for a job for which the load and Sensible Heat Factor are known:

$$q = \frac{Q_s}{1.08(1\text{-BF})\ (t_r - t_s)} \quad \ldots\ldots\ldots\ldots \quad (3)$$

where

q = flow of air, cfm
Q_s = sensible heat load, Btu per hr
BF = bypass factor
t_r = room temperature
t_s = apparatus dew point

COIL PERFORMANCE, OVERALL

Since with all finned coil dehumidifiers there is a substantial difference between the coil *apparatus dew point* and the refrigerant temperature, it is essential to know the refrigerant side performance as well as the air side. Several methods of analyzing coil tests[11] and extending data to other coil conditions have been developed. The *total heat* or *three line* method developed by the author and by others[7] determines the terminal surface conditions for any chosen air and refrigerant conditions. This method (Fig. 6) is based upon determining the terminal surface conditions by making the ratio:

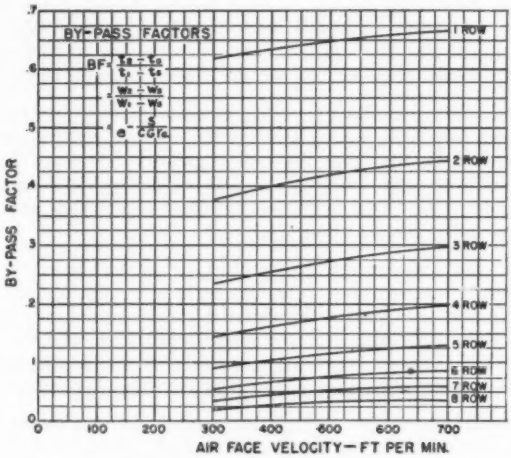

FIG. 5. TYPICAL BYPASS FACTOR FOR A LINE OF FINNED COILS

$$\frac{\text{total heat difference between air and surface}}{\text{temperature difference between surface and refrigerant}}$$

equal to

$$\frac{\text{heat transfer resistance of air film} \times \text{specific heat of moist air}}{\text{heat transfer resistance of refrigerant film and metal}}$$

From these data, the depth of the coil, and, hence, the overall coil performance for the assigned conditions are determined. This basic method has been further refined[8].

The refrigerant side heat transfer is expressed in terms of the temperature difference between the coil apparatus dew point calculated from the Bypass Factor (Equation 2) and the refrigerant temperature calculated as described previously. It has been found that this refrigerant side performance is relatively independent of the air side performance. For a given coil, using a non-volatile refrigerant, it may be expressed in terms of load and refrigerant flow[11]; for a volatile refrigerant, in terms of load[9]. (Illustrated in Table 1.)

PROBLEMS DUE TO OUTSIDE AIR

As far as possible, outside air for ventilation is brought through the apparatus, to reduce to a minimum the direct effect of outside air on the room conditions. This also increases the effectiveness of the air conditioning apparatus, because the entering air is increased in temperature and moisture content above the room conditions.

The ventilation requirements are determined by the sources of air contamination: people, food, etc. Thus, the outside ventilation air for a given job

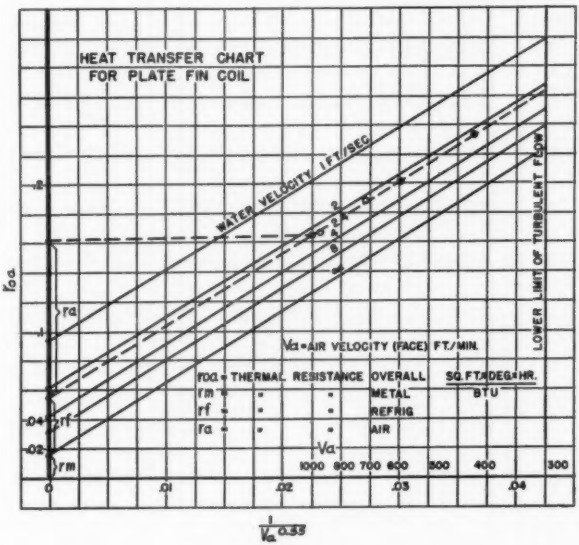

FIG. 6. HEAT TRANSFER CHART FOR PLATE FIN COIL

is chosen as a certain flow rate in cubic feet per minute. But the dehumidified air quantity is determined from the air conditions leaving the apparatus and depends upon the room load and the Sensible Heat Factor. Where a saturating type of dehumidifier is used, determining the condition of the air leaving the apparatus is simple, since the air is at saturation and is not affected by the condition of the air entering the coil.

On the other hand, where a coil dehumidifier is used which does not saturate the air, the air condition leaving the coil is a function not only of the effective coil surface temperature and of the coil Bypass Factor, but also of the entering air condition. But the entering air condition is unknown until the air quantity passing through the coil is chosen and the *percentage* of outside air can be determined.

TABLE 1. DIRECT EXPANSION COOLING

GRAND TOTAL HEAT (Btu/hr in Thousands)						REFRIGERANT TEMPERATURES (F)													
COIL FACE AREA (SQ FT)						APPARATUS DEW POINT (F)													
2.25	4.5	6.75	9	13.5	18														
cubic feet per minute																			
1000	2000	3000	4000	6000	8000	38	40	42	44	46	48	50	52	54	56	58	60	62	64
4 ROWS—BYPASS FACTOR 0.17																			
30.0	59.6	90.5	122.5	182.0	242.0	*27*	*29*	*31*	33	35	37	39	41	43	45	47	49	51	53
32.0	63.6	95.0	131.0	194.0	258.0	*26*	*28*	*30*	32	34	36	39	41	43	45	47	49	51	53
34.0	67.6	103.0	139.0	206.0	274.0	*26*	*28*	*30*	32	34	36	38	40	42	45	47	49	51	53
36.0	71.5	109.0	147.0	218.0	290.0	*26*	*28*	*30*	32	34	36	38	40	42	44	47	49	51	53
38.0	75.5	115.0	155.0	231.0	306.0	*26*	*28*	*30*	32	34	36	38	40	42	44	46	48	51	53
40.0	79.5	121.0	163.0	243.0	322.0	*25*	*27*	*29*	*31*	33	36	38	40	42	44	46	48	50	53
42.0	83.5	127.0	171.5	255.0	338.0		*27*	*29*	*31*	33	35	38	40	42	44	46	48	50	52
44.0	87.4	133.0	179.5	267.0	355.0		*27*	*29*	*31*	33	35	37	40	42	44	46	48	50	52
46.0	91.4	139.0	188.0	279.0	371.0		*26*	*28*	*30*	33	35	37	39	41	44	46	48	50	52
48.0	95.5	145.0	196.0	291.0	387.0		*26*	*28*	*30*	32	34	37	39	41	43	45	48	50	52
50.0	99.4	151.0	204.0	303.0	404.0		*25*	*27*	*30*	32	34	36	38	41	43	45	47	49	52
52.0	103.2	157.0	212.0	316.0	419.0			*27*	*29*	*31*	34	36	38	40	43	45	47	49	51
54.0	107.2	163.0	220.0	328.0	436.0			*26*	*28*	*31*	33	35	38	40	42	44	47	49	51
56.0	111.0	169.0	228.0	340.0	452.0			*26*	*28*	*30*	33	35	37	39	42	44	46	48	51
58.0	115.5	176.0	237.0	352.0	468.0			*25*	*27*	*30*	32	34	37	39	42	44	46	48	50
60.0	119.0	181.0	245.0	364.0	484.0				*26*	*29*	*31*	34	36	38	41	43	45	48	50
62.0	123.2	187.0	253.0	376.0	500.0				*26*	*28*	*31*	33	36	38	40	43	45	47	49
64.0	127.0	193.0	260.0	388.0	516.0				*25*	*28*	*30*	33	35	38	40	42	45	47	49
66.0	131.0	199.0	268.0	400.0	532.0					*27*	*29*	32	35	37	39	42	44	46	49
68.0	135.2	205.0	277.0	413.0	548.0					*26*	*28*	*31*	34	36	39	41	43	46	49
70.0	139.2	212.0	285.0	425.0	565.0					*25*	*27*	*30*	33	35	38	40	43	45	49
6 ROWS—BYPASS FACTOR 0.07																			
36.0	71.5	109.0	147.0	218.0	290.0	*30*	32	34	36	38	40	42	44	46	48	50	52	54	56
38.0	75.5	115.0	155.0	231.0	306.0	*29*	*31*	33	36	38	40	42	44	46	48	50	52	54	56
40.0	79.5	121.0	163.0	243.0	322.0	*29*	*31*	33	35	38	40	42	44	46	48	50	52	54	56
42.0	83.5	127.0	171.5	255.0	338.0	*29*	*31*	33	35	37	39	41	44	46	48	50	52	54	56
44.0	87.4	133.0	179.5	267.0	355.0	*28*	*31*	33	35	37	39	41	43	45	48	50	52	54	56
46.0	91.4	139.0	188.0	279.0	371.0	*28*	*30*	32	35	37	39	41	43	45	47	49	52	54	56
48.0	95.5	145.0	196.0	291.0	387.0	*28*	*30*	32	34	36	39	41	43	45	47	49	51	53	56
50.0	99.4	151.0	204.0	303.0	404.0	*27*	*29*	32	34	36	38	40	43	45	47	49	51	53	55
52.0	103.2	157.0	212.0	316.0	419.0	*27*	*29*	*31*	34	36	38	40	42	44	47	49	51	53	55
54.0	107.2	163.0	220.0	328.0	436.0	*26*	*29*	*31*	33	35	38	40	42	44	46	49	51	53	55
56.0	111.0	169.0	228.0	340.0	452.0	*25*	*28*	*30*	33	35	37	39	42	44	46	48	50	53	55
58.0	115.5	176.0	237.0	352.0	468.0		*27*	*30*	32	35	37	39	41	43	46	48	50	52	55
60.0	119.0	181.0	245.0	364.0	484.0		*27*	*29*	32	34	36	39	41	43	45	48	50	52	54
62.0	123.2	187.0	253.0	376.0	500.0		*26*	*29*	*31*	33	36	38	40	43	45	47	50	52	54
64.0	127.0	193.0	260.0	388.0	516.0		*25*	*28*	*30*	33	35	38	40	42	45	47	49	52	54
66.0	131.0	199.0	268.0	400.0	532.0		*25*	*27*	*30*	32	35	37	39	42	44	47	49	51	54
68.0	135.2	205.0	277.0	413.0	548.0			*26*	*29*	*31*	34	36	39	41	44	46	49	51	53
70.0	139.2	212.0	285.0	425.0	565.0			*25*	*28*	*30*	33	36	38	41	43	46	48	51	53

From this dilemma there are three possible avenues of escape which do not involve cut-and-try or approximation: (1) choose the air velocity and depth, *i.e.*, the *Bypass Factor*, of the dehumidifying coil; (2) choose the dehumidified air quantity; and (3) choose the leaving air conditions. The choice which is

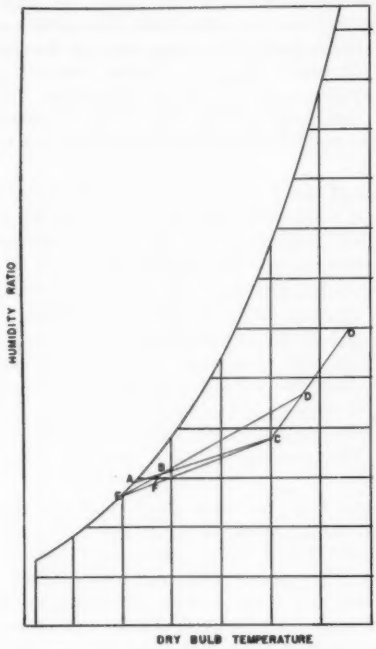

FIG. 7. RELATION OF OUTSIDE AIR LOAD AND BYPASSED AIR LOAD ON PSYCHROMETRIC CHART

made is perhaps the most significant difference between estimating methods, as will be shown in a later section.

For the method developed here, the cooling coil is chosen as to air velocity and rows deep, as a part of the heat load calculation. The choice is simple and natural, because cooling coils are usually available in integral row increments of depth. This choice is based upon economic factors which are related to the type of job and type of load and which are readily learned by the estimator.

Since a portion of the air entering the coil is assumed to pass through without change of condition, this portion of the outside air can be considered as *bypassed* through the coil. The effect on the room load is the same as if this outside air entered the room independently of the apparatus, as infiltration air

does. Thus, it is evident that the load due to the *bypassed* outside air can be treated in the same manner as infiltration load and therefore added to the room load.

The *dehumidified* portion of the outside air leaves at saturation, and its effect on the room is as though it were an equal amount of air from the room. Thus the new room sensible and latent heats (including the heat from the *bypassed* portion of the outside air) serve as the basis of a pseudo room Sensible Heat Factor line on which the coil apparatus dew point is located. The Bypass Factor, when applied to the room temperature minus the coil apparatus dew point, gives a pseudo air temperature leaving the apparatus. When used with the room sensible heat (including sensible heat from *bypassed* outside air) this pseudo air temperature permits the calculation of the dehumidified air quantity according to Equation 3.

The outside air load of the *dehumidified* or non-*bypassed* outside air must, of course, be added to the room total heat to give the grand total heat. It should be noted that, when this method is used, it is not necessary to determine the actual air conditions entering and leaving the coil.

The relationships described can be seen graphically on the psychrometric chart, Fig. 7. The slope of the line AC represents the Sensible Heat Factor for the room condition at C and A the apparatus dew point for the room. Let O be the outside air condition and D on line CO the condition of the mixed air entering the coil. Assume the condition of the air leaving the coil as B on line AC. Then the coil apparatus dew point must be at E where the extension of the line DB intersects the saturation line and this must be the *surface temperature* of the coil. This corresponds to a room Sensible Heat Factor represented by the line EC.

If now a line is drawn from the leaving coil condition B parallel to CO meeting line EC at F, two similar triangles EBF and EDC are formed. Therefore, $EB/ED = BF/DC = EF/EC$ = coil Bypass Factor (equivalent sides of similar triangles). But since DC represents the outside air load, BF must represent the *bypassed* portion of the outside air load. This can be attached to BC representing the room load to give a new equivalent or pseudo room load represented by FC. By means of the transfer to the room load of the *bypassed* portion of the outside air load (represented by BF) a new value can be determined for room Sensible Heat Factor (represented by line FC). Thus the coil apparatus dew point E becomes the pseudo room apparatus dew point. The air quantity passing through the coil may thus be figured from the coil apparatus dew point E, the room condition C, and the coil Bypass Factor EF/EC.

CALCULATING AIR CONDITIONING ESTIMATE

A typical cooling load estimate made in accordance with the principles outlined in this paper is shown on the cooling estimate form (Fig. 1). The highlights of the estimate are: (1) pre-selection of the outside and room conditions; (2) calculation of the ventilation requirements in terms of air quantity; (3) determination of the *bypass* outside air load using the coil *Bypass Factor* determined from the pre-selected coil depth and face velocity; (4) determination of the Sensible Heat Factor and from this, the coil apparatus dew point; (5) calculation of the dehumidified air quantity based upon the room condi-

tions, the coil apparatus dew point and the coil *Bypass Factor;* and (6) selection of the equipment as determined by the grand total heat and the refrigerant temperature.

COMPARISON OF METHODS

Most load estimating and equipment selection methods which are designed to produce a guaranteed result recognize the need for breaking down the load into its sensible and latent components. These methods use some equiva-

TABLE 2. COMPARISON OF DIFFERENT SYSTEMS OF LOAD ESTIMATING AND COOLING COIL SELECTION

ITEM	SYSTEMS				
	1[c]	2	3	4	5
Outside Air, cfm	Given	Given	Given	Given	Given
Outside Air DB	Given	Given	Given	Given	Given
Outside Air Moist Cont.	Given	Given	Given	Given	Given
Room DB	Given	Given	Given	Given	Given[a]
Room Moist Cont.	Given	Given	Given	Given[a]	Given[a]
Room Sensible Heat	Calculate[b]	Calculate	Calculate	Calculate	Calculate[a]
Room Latent Heat	Calculate[b]	Calculate	Calculate	Calculate[a]	Calculate[a]
Outside Air Sens Heat	Calculate[b]	Calculate	Calculate	Calculate	Calculate[a]
Outside Air Latent Heat	Calculate[b]	Calculate	Calculate	Calculate[a]	Calculate[a]
Sensible Heat Factor	Calculate	Calculate	Calculate	Calculate[a]	Calculate[a]
Apparatus Dew Point	Result 2	Result 5	Result 5	Assume[a]	Assume[a]
Apparatus cfm	Result 3	Assume	Result 1	Result 1	Result 1
Apparatus DB	—	Result 1	Assume	Assume	Assume
Apparatus Moist Cont.	—	Result 2	Result 2	Assume[a]	Assume
Outside Air & Room Mixture DB	—	Result 3	Result 3	Result 2	Result 2
Outside Air & Room Mixture Moist Cont.	—	Result 4	Result 4	Result 3	Result 3
Coil Velocity	Assume	Assume	Assume	Assume	Assume
Coil Depth	Assume	Result 7	Result 7	Assume	Result 4
Coil Bypass Factor	Result 1	Result 6	Result 6	Assume[a]	Result 5
Refrigeration Temp.	Result 4	Result 8	Result 8	Result 4	Assume

Notes: Numbers indicate normal sequence of resultant steps.
[a] Approximate only.
[b] *Bypassed* outside air heat included in room heat.
[c] No. 1 system is that described in this paper.

lent of the Sensible Heat Factor for determining the state point of the air leaving the apparatus. Some of these systems also recognize the principle of the coil apparatus dew point and of the Bypass Factor.

The principle differences occur in the determination of the dehumidified air quantity and the selection of the cooling coil. Table 2 illustrates in chart form the salient differences between the method presented in this paper and other possible means of treating the problem.

Systems 1, 2 and 3 represent the three solutions listed under *Problems Due to the Outside Air.* In System 4, the coil depth and the coil leaving air conditions are assumed, whereas in System 5 the refrigerant temperature and the coil leaving air conditions are assumed. Both systems may result in substantial deviations, from design, of the room humidity and of the latent heat load. The numbers following the results indicate the probable order of the steps and provide some index to the calculation.

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SYMBOLS

A = face area of the coil, square feet

(BF) = bypass factor

c = specific heat of moist air, Btu per pound. (dry air)

G = weight flow of air, pound per hour. (dry air)

h = Enthalpy of moist air, Btu per pound. (dry air)

K = Mass transfer coefficient of moisture through air film, pounds per square foot × hours × pounds per pound

L = coil characteristic ratio. (See Equation 30)

M = moisture transfer, pounds per hour

q = volume flow of air, cubic feet per minute. (Standard conditions)

Q_L = latent heat, Btu per hour

Q_s = sensible heat, Btu per hour

Q_T = total heat, Btu per hour

r_a = heat flow resistance of air film, hours × square feet × degree per Btu

r_m = heat flow resistance of metal, hours × square feet × degree per Btu

r_f = heat flow resistance of refrigerant film, hours × square feet × degree per Btu

S = surface area of the coil (air side), square feet

(SHF) = sensible heat factor

t = temperature, Fahrenheit degree

t_s = coil apparatus dew point

V_a = velocity of the air over the coil face, feet per minute

W = humidity ratio, pounds of moisture per pound of dry air

ρ = density, pounds per cubic foot

SUBSCRIPTS

$_1$ = entering condition
$_2$ = leaving condition
$_r$ = room condition
$_s$ = surface condition
$_w$ = moisture condition
$_f$ = refrigerant condition

APPENDIX I

DERIVATION OF SENSIBLE HEAT, LATENT HEAT, TOTAL HEAT AND SENSIBLE HEAT FACTOR

The enthalpy of moist air, h, may be expressed within the limits of 32 to 100 F with an accuracy of 0.10 percent by the equation:

$$h = (0.2402 + 0.44W)t + 1061W \quad \ldots\ldots\ldots\ldots (4)$$

In a process involving the removal of heat and moisture from air, the heat removed per unit weight of dry air is equal to the difference of the enthalpies corresponding to the entering and leaving conditions of the air minus the enthalpy of liquid water condensed from a unit weight of dry air. For processes above 32 F, this heat difference may be expressed as:

$$\frac{Q_T}{G} = h_1 - h_2 - (t_{w2} - 32)(W_1 - W_2) \quad \ldots\ldots\ldots\ldots (5)$$

Substituting Equation 4 in Equation 5 and re-arranging:

$$\frac{Q_T}{G} = (0.2402 + 0.44W_2)(t_1 - t_2) + (1093 - t_{w2} + 0.44t_1)(W_1 - W_2) \quad \ldots (6)$$

It is possible to simplify Equation 6 by assuming certain specific values for some of the terms of second order value. Substitute representative values for these second order terms as follows:

W_2 = 0.0087 lb moisture per pound of dry air (53½ F dew point)
t_1 = 80 Fahrenheit
t_{w2} = 52 Fahrenheit

then

$$\frac{Q_T}{G} = 0.244(t_1 - t_2) + 1076(W_1 - W_2) \quad \ldots\ldots\ldots\ldots (7)$$

If other values of W_2, t_1 and t_{w2} are substituted in Equation 6 and compared with Equation 7, then Equation 7 can be used in solving the normal problem of cooling and removing moisture from room air for comfort conditioning purposes, with an error varying from 0 to ½ percent; whereas the same equation can be used for determining the removal of heat and moisture from outside air with an error of ½ to 1 percent.

The first term on the right side of Equation 7 may be described as the *sensible heat*, either based on a unit weight or on the pounds per hour of air used in the process, as in Equation 8. Or it may be expressed in terms of the volume of standard air in cubic feet per minute as in Equation 8*a*:

$$Q_S = 0.244G(t_1 - t_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

$$Q_S = 1.08q(t_1 - t_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots (8a)$$

Note that the specific volume of standard air (0.075 pounds per cubic foot) must be multiplied by the volume ratio of dry to moist air at the assumed condition $\left(1 - \frac{0.0087}{0.62}\right) = 0.986$ to get the specific volume of dry air at standard conditions. Similarly, the second term of Equation 7 may be described as *latent heat* and may be expressed either in terms of unit-weight or of pounds of air per hour as in Equation 9, and may also be expressed in terms of standard air (cfm) as in Equation 9*a*:

$$Q_L = 1076G\,(W_1 - W_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (9)$$

$$Q_L = [0.68q]\,[7000\,(W_1 - W_2)] \quad \ldots\ldots\ldots\ldots\ldots \quad (9a)$$

Note that the multiplying factors in Equations 8*a* and 9*a* are those used in connection with the air conditioning estimate for sensible heat and latent heat respectively; also, that the second bracketed term of Equation 9*a* includes the conversion factor from pounds to grains. If W were in grains per pound, the 7000 would be cancelled out. Q_T, the total heat, may then be expressed as the sum of the sensible and latent heat:

$$Q_T = Q_S + Q_L \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

While the terms *sensible, latent,* and *total heat* do not have an absolute physical significance, they are, in practice, very significant to the air conditioning engineer, and their use greatly simplifies the mechanics of load computation.

Sensible Heat Factor is defined as the ratio of the sensible to the total heat:

$$(\text{SHF}) = \frac{Q_S}{Q_T} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

By substituting Equations 8 and 9 in Equation 11, Sensible Heat Factor may be expressed in terms of temperature difference and humidity ratio difference:

$$(\text{SHF}) = \frac{0.244\,(t_1 - t_2)}{0.244\,(t_1 - t_2) + 1076\,(W_1 - W_2)} \quad \ldots\ldots\ldots \quad (12)$$

This expression can be simplified further to show the humidity ratio difference to the temperature difference as a simple ratio:

$$(\text{SHF}) = \frac{1}{1 + 4400\,\dfrac{(W_1 - W_2)}{(t_1 - t_2)}} \quad \ldots\ldots\ldots\ldots\ldots \quad (12a)$$

Equation 12*a* provides the information necessary to plot the slope of the condition line corresponding to any sensible heat factor on the psychrometric chart. Note that W in Equation 12*a* is in pounds per pound.

APPENDIX II

DERIVATION OF BYPASS FACTOR

The sensible cooling capacity Q_S of a coil may be stated in terms of its surface, air film resistance, and log mean temperature difference as:

$$Q_S = \frac{S}{r_a} \times \frac{(t_1 - t_s) - (t_2 - t_s)}{\log_e\left(\dfrac{t_1 - t_s}{t_2 - t_s}\right)} \quad \ldots\ldots\ldots\ldots\ldots \quad (13)$$

where t_s is assumed to be constant. But Q_S can also be expressed as:

$$Q_S = Gc(t_1 - t_2) = 60 V_a A \rho c (t_1 - t_2) \quad \ldots \ldots \ldots \quad (14)$$

Combining Equations 13 and 14 and simplifying:

$$Gc = \frac{S}{r_a} \times \frac{1}{\log_e \left(\frac{t_1 - t_s}{t_2 - t_s} \right)} \quad \ldots \ldots \ldots \ldots \quad (15)$$

This may be converted from the log form to a power of e in terms of the Bypass Factor (BF):

$$BF = \frac{t_2 - t_s}{t_1 - t_s} = e^{-\frac{S}{Gcr_a}} = e^{-\frac{S}{60 V_a \rho c r_a}} \quad \ldots \ldots \ldots \quad (16)$$

Lewis[5] has developed in approximate form the relationship of the heat transfer from air to surface and the corresponding mass transfer of moisture as:

$$cr_a K = 1 \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (17)$$

In a manner similar to Equation 13 for heat transfer, the mass transfer of moisture to the coil can be written:

$$M = SK \times \frac{(W_1 - W_s) \times (W_2 - W_s)}{\log_e \frac{W_1 - W_s}{W_2 - W_s}} \quad \ldots \ldots \ldots \quad (18)$$

where

W_s is assumed constant

likewise:

$$M = G(W_1 - W_2) = 60 V_a A \rho (W_1 - W_2) \quad \ldots \ldots \ldots \quad (19)$$

Combining Equations 17, 18 and 19 and recasting as in Equations 15 and 16:

$$BF = \frac{W_2 - W_s}{W_1 - W_s} = e^{-\frac{S}{Gcr_a}} = e^{-\frac{S}{60 V_a \rho c r_a}} \quad \ldots \ldots \ldots \quad (20)$$

In the given equations t_s and W_s were assumed constant but Equations 13 and 14 can also be written:

$$\sum_1^2 \Delta Q_s = r_a \sum_1^2 (t - t_s) \Delta S = Gc(t_1 - t_2) \quad \ldots \ldots \quad (21)$$

where values of $(t - t_s)$ are the actual values of air to surface temperature difference through the coil as shown in Fig. 5. From Equations 13 and 21 the mean temperature difference is:

$$\frac{(t_1 - t_s) - (t_2 - t_s)}{\log_e \left(\frac{t_1 - t_s}{t_2 - t_s} \right)} = \frac{1}{S} \sum_1^2 (t - t_s) \Delta S \quad \ldots \ldots \ldots \quad (22)$$

It is apparent that the left-hand term of Equation 22 is a particular form of the right-hand term in which t_s is given a value to satisfy Equation 13. Thus, the significance of t_s or coil apparatus dew point is that it is the constant surface temperature necessary to produce the same results as were actually produced with the variable surface temperature. The same reasoning applies to the mass transfer of

moisture, and the constant surface condition can be considered the same for both heat and moisture transfer.

DISCUSSION

L. T. AVERY, Cleveland, Ohio: I want to thank Mr. Ashley for this paper on applied air conditioning. We have had letters of criticism from our more erudite members, that we have too much practical applied science. In the same mail we get letters from others who say that the meetings are entirely too high-brow, too technical, too philosophical; and that we need more applied science. So we mix up our meetings. This is a good paper for the applied science advocates.

There are three points I would like to emphasize because Mr. Ashley has brought them out. We still sin in the respect of neglecting to bring in outside air through the apparatus. In spite of the fact this requirement has been known and practiced for 15 years, we still find thousands of dollars spent on apparatus which cannot possibly do the job because of this one design feature.

The depth of the coil has been overemphasized to get additional cooling. In the old days, we thought in terms of getting complete saturation in order to simulate the washer. The cleanliness of the coil is an important problem. The greater the depth of that coil, the more difficult the problem of cleaning. Therefore, you strike a compromise in order to have a practical coil that can be cleaned. The design guarantee that Mr. Ashley mentioned so lightly will lead you astray. The old design of 80 deg and 50 percent relative humidity or 74 effective, we find entirely too high, and systems are now designed for 71 deg effective. They must be carried at 76 to 77 dry-bulb and 45 to 50 percent relative humidity, if you wish to do a satisfactory job. So we have at least three features of applied science that should help those of you who are new in this business. These should also help those of us who are more experienced, but who have been careless.

P. R. ACHENBACH, Washington, D. C.: I have two questions regarding the figures in this paper. One is whether the bypass factor of the specific coil is a function of the coil only, or if it is not also a function of the sensible heat appearing in the room? I believe, as he defined his term *bypass factor*, it implied that it was a function only of the coil.

Secondly, referring to Fig. 7, in which a graphic method for determining the effect of outdoor air was used, I question whether that method provides a means for preselecting the difference between the supply air temperature and the room dry bulb that you wish to maintain.

L. G. SEIGEL, Cleveland, Ohio: Mr. Ashley is to be complimented upon presenting methods of solution of so important a problem. I would like to raise one point. Inasmuch as the method of solution proposed seems to depend primarily upon the knowledge of the apparatus dew point, which is determined by the intersection of the load-ratio line with the saturation curve, I would like to ask whether problems without such intersection can be solved by the method proposed by Mr. Ashley?

F. EDWARD INCE, St. Louis, Mo.: Mr. Ashley, in Fig. 4 you show the coil performance line as being a curved line intersecting the room performance line. We know that to be a fact from test data. However, the extreme bending of this line is a function of refrigerant temperature as well as the loading of the coil. Since the intersection point would be a function of the bypass, I feel that this point would be a little elusive to locate. True enough, we should recognize this bypass if it actually occurs. However, assuming that the mixed air leaving a coil is at given dry bulb and wet bulb temperatures, why is it that we cannot design our supply air requirements based on these figures? Why add the bypassed air load to the space?

The subject of bypassing outside air should be investigated further, since many authorities advise against this practice. While this remark does not pertain to this paper, it should be pointed out that in many present day applications recirculated and fresh air mixtures are bypassed.

S. P. GOETHE, Gainesville, Fla. (WRITTEN): The method presented by Mr. Ashley is rational and the comprehensive discussion of Fig. 7 illustrates pseudo room apparatus dew point. There are, however, a few comments which this writer thinks might be of interest to the readers of Mr. Ashley's paper.

1. It has been the writer's experience that the air leaving spray type dehumidifier (commercially manufactured units) is never saturated, but always has a bypass factor.

2. According to many manufacturers' performance data, the line CDB of Fig. 4 is a straight process line, providing the initial surface of the evaporator is wet. This condition generally exists provided reasonable face velocity and refrigerant evaporator temperature are used.

3. In Equation 3, the definition of term Q_s is incomplete. When this equation is used as written, Q_s is used to denote the total sensible heat as shown in the glossary of terms T_r (room dry bulb temperature), Equation 4 must be replaced by the dry bulb temperature of the air mixture entering the evaporator.

4. It is the writer's opinion, which is evidently shared by the authors of Systems 2, 3, 4 and 5 as shown in Table 2, that the bypass concepts as explained in Paragraph 0, page 000, is not an exact statement of fact, but rather a concept of utility. It is this writer's contention that the room heat gain is independent of the evaporator depth. The heat quantities shown for infiltration should be included not as part of the heat gain, but rather as a portion of the outside-air heat gain, unless Mr. Ashley's method of evaporator selection is used. Further, it should be noted that the true cycle of the air conditioning system is Triangle DBC of Fig. 7.

5. The inclusion of the bypass concept in the heat gain estimate automatically fixes the number of rows of evaporator depth, leaving, as variables for evaporator selection, the refrigerant evaporating temperatures and the face area. Usually the refrigerant evaporator temperature range is limited due to economic considerations; therefore, the only convenient variable remaining is the evaporator face area. This writer prefers to select a reasonable leaving air condition from the evaporator, and thereby have more latitude and selection of the evaporator, that is, varying the number of rows of depth and the face area without necessity of adjusting the heat gain estimate.

AUTHOR'S CLOSURE: With respect to the points raised by Mr. Achenbach, the bending of the condition line, Fig. 4, is shown somewhat extreme for the purpose of illustration. The condition line with constant refrigerant temperature is usually bent less than is shown. The pre-selection of the difference between the supply air temperature and the room dry bulb is shown in Table 2 as Selection System 3 which is possible but not recommended because of its greater complexity. In the practical application of the recommended method (*i.e.*, pre-selecting the coil depth), it is possible to make a reasonable approximation of any desired supply-air to room temperature difference after having developed a little experience in the use of the method.

With respect to the question raised by Mr. Seigel concerning a case where the extension of the line between the entering and leaving conditions does not intersect the saturation line—if such a condition exists, it represents a deviation from the Lewis Equation. While such a deviation is recognized as possible, the Lewis Equation represents a more conservative expression of the coil performance and is, therefore, recommended for use until such time as a rational alternative relationship is available. In either case, it is possible to use the bypass method since, if the mass transfer is not proportional to the heat transfer, a different bypass factor can be used for moisture and heat removal.

With reference to the comments of Mr. Ince, the bypass is not at all illusive, but is very tangible and can be evaluated as shown in the paper. On a shallow coil, a considerable error can be made in the room conditions if the outside air, which is bypassed through the coil, is neglected.

With respect to Prof. Goethe's comments:

It is true that some of the shorter spray dehumidifiers used today do not fully saturate the air. Where this is the case, essentially the same problem exists as with surface type dehumidifiers.

The condition line of Fig. 4 cannot be a straight line except where the surface temperature of the coil is constant with depth. This almost never occurs on extended surface dehumidifiers.

There is some possibility of confusion with respect to Equation 3 which would be eliminated by the use of t_1 instead of t_r thus making it clear that the calculation of load pertains to the coil rather than to the room.

Systems 2, 3, 4 and 5 were chosen as representing preferred alternate choices and not as actual alternate systems in commercial use, although it is possible that some of them may be. As shown in the paper, within the limits of the assumptions made, the bypass concept is an exact statement of fact; however, the problem of the room load may be approached from several different points of view as shown in the alternate systems. Implicit in all of them, however, is the fundamental bypass concept. Whether the bypassed outside air should properly be included as a part of the room or outside air load depends entirely upon whether systems 1 or systems 2, 3, 4, or 5 are used.

If, as the discusser points out, he assumes a leaving air condition from the evaporator and varies the number of rows of coil depth, the result will be to vary the effective apparatus dew point and thus the room relative humidity and the latent heat load on the system. Where specific guarantees are to be met, this is an undesirable procedure. Actually, if it is desired to make a second assumption as to the depth of the coil with the proposed method of estimating, this can be done with a relatively small amount of change in the estimate. In practice, it is seldom necessary because the type of job and the available coil sizes and depths help to guide the correct initial choice.

1355

No. 1355

ARE AUTOMATIC AIR SHUTTERS JUSTIFIED ON A GAS-FIRED CONVERSION BURNER?

By William M. Myler, Jr.* and Harlan W. Nelson,** Columpus, Ohio

Since the earliest days of gas-fired conversion equipment, burners have been offered to the public both with and without automatic air shutters which were designed to close off the flow of air for combustion during the period when the burner was not in operation. Whether these automatically controlled shutters increased operating efficiency, and thereby decreased operating costs, has been argued pro and con for years. Although from time to time tests have been run trying to determine the facts about the control of *off-period* air, a search of the literature revealed no reports of comprehensive and authoritative tests.

This investigation, therefore, was conducted for the purpose of obtaining data from which the value of automatic air shutters might be determined.

Problems of Shutter Design

At first thought, it seems like a very simple matter to provide a small door or shutter which opens and closes each time the gas valve opens and closes on a conversion burner. Experience has proved otherwise. It should be clearly understood that a closure which would have zero air leakage is not being considered.

The power available for operating such a door is small. It is usually supplied by a valve arm extending from an electrically operated valve, or from a small gas-actuated diaphragm. Such a system must operate reliably down to the minimum gas pressure at which the burner will function satisfactorily. The system consisting of the door and its linkage mechanism, must be so reliable that it will operate throughout the life of the appliance which may be as much as 20 years. The cycles of operation during that period may number a total of several hundred thousand.

The system must be rugged enough to withstand rough handling by installation and service men. It must be simple to adjust and must positively maintain that adjustment. The linkage mechanism must be so designed that, if it breaks

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or becomes disconnected, the door will fall open. Conversely, if the door gets jammed open and cannot close, the valve must still be able to close properly. If the damper remains closed for any reason, the gas valve must be prevented from opening. Last, but not least, it must be inexpensive to manufacture.

GENERAL PLAN OF TESTS

For test purposes, a conversion burner was installed in a hot water boiler of the wet-base type. The boiler had a required input of approximately 140,000 Btu per hr. The burner was fired intermittently or continuously in a program involving several variables. The variables were (1) the length of the operating cycle, (2) the percentage of *on* operation within each cycle, and (3) the position of the air shutter during *off-periods,* whether open or closed. In addition to the intermittent runs, several continuous runs were made with the excess air maintained at different percentages. This was achieved by opening the air shutter to different percentages.

The *A.G.A.* method of determination of hot-water boiler efficiency could not be used on these tests. In the *A.G.A.* method, water is introduced into the boiler at room temperature and the rate of flow through the boiler is adjusted to produce a 180 F outlet water temperature. The output water is weighed. This test condition is not comparable to a normal operating condition of a hot-water boiler. In the *A.G.A.* method, the cold water addition during the *off-period* on a cycling test would materially affect the flow of air through the boiler by reducing boiler temperature too rapidly.

For this reason, it was decided to set up the boiler as a gravity circulating system. The boiler was connected to a copper tube heat exchanger located at a height above the boiler simulating the height of first-floor radiation above a boiler installed in a basement. Heat was dissipated from the exchanger by cooling water circulated through it by a small pump. The system was so adjusted that when the boiler was being fired continuously at rating (140,000 Btu per hr), the outlet water temperature was about 190 F and the return temperature 160 F. By maintaining the same rate of cooling water flow for intermittent tests as that established for continuous operation, a condition was obtained comparable to that in a residence.

In determining the test procedures to be followed, it was recognized that quite accurate measurements of the several factors involved in the tests would be required because it seemed probable that the magnitude of the variation in efficiency to be obtained would not be very large.

CONCLUSIONS FROM TEST DATA

The conclusions to be drawn from the data presented in this report can be summarized as follows:

1. The relationship between the *off-period* air loss with open or closed shutters, and the percentage of *on-period* time during cyclic operation, has been successfully established by direct measurement of the rate of flow and rise in temperature of the air passing through the boiler during *off-periods.*

2. The loss in sensible heat of the *off-period* air was found to decrease gradually as the percentage of *on* time within the cycle was increased. The loss at 20 percent *on* operation, expressed as percentage of total heat input, averaged 4.4 percent with

the air shutters open and decreased to 1.8 percent with the shutters closed, a difference of 2.6 percent. At 66⅔ percent *on* time, the losses with the air shutters open and closed were 1.3 and 0.7 percent, respectively, a difference of 0.6 percent.

3. The ability to determine the effect of the position of the air shutters on the overall boiler efficiency was found to be limited by the precision with which results could be duplicated for comparable tests. Excepting in results from one test, the boiler efficiencies for successive tests under identical conditions deviated from the average for each series by amounts ranging from ±0.3 percent to ±0.9 percent.

4. The overall boiler efficiency, expressed as percentage of heat input, was found, as expected, to increase as the percentage of *on* time was increased. Except for tests with 50 percent *on* time, the magnitude of the difference between the boiler efficiencies with the air shutters open and with them closed decreased as the percentage of time *on* became greater. The results for the closed shutter tests with 50 percent *on* time are out of line with those of all other tests. Examination of the data for these tests, and comparison with results from other tests have not revealed the cause for the discordant results.

5. Disregarding the results of the tests at 50 percent *on* time, which (as pointed out in conclusion 4) are out of line, there is excellent agreement between the *average* difference in thermal efficiency with air shutters open and closed during the *off-period* and the *average* difference in the heat loss of the *off-period* air with air shutters open and closed for the same series of tests as shown in the following table.

PERCENTAGE *ON* TIME	PERCENT DIFFERENCE IN THERMAL EFFICIENCY	PERCENT DIFFERENCE IN *OFF*-PERIOD AIR LOSS
20	2.2	2.6
33⅓	1.4	1.5
66⅔	0.3	0.6

6. If the differences in heat loss in the *off-period* air are weighted by the percentage of *on* time, it is found that the savings in heat from use of shutters remain almost constant when referred to the rated input of the heater. Thus $2.6\times0.20=0.52$, and $0.6\times0.666=0.40$.

Significance of the Conclusions Based on the Data: Although the data presented show that thermal efficiency is increased when *off-period* air is controlled by an automatic shutter, the more important consideration is the magnitude of the increase in efficiency. This efficiency is of concern primarily as it affects seasonal operating cost.

Taking as a typical example a house in Columbus, Ohio, that could be heated with the conversion burner used in these tests, such a well built seven room house can be heated with natural gas for approximately $100 for the entire heating season.

For the purpose of a sample calculation, let it be assumed that it costs exactly $100 to heat this house in a year when the degree-days are normal. In the last eight years, the degree-days in Columbus have varied from 89 percent to 107 percent of normal, which means that the operating costs for this house varied between $89 and $107.

The heating value of the natural gas in Columbus varies as much as 5 percent, depending upon the source of supply. That accounts for another $5 variable in the annual operating cost.

Legally, a gas meter is only required to be within 3 percent of accuracy, thereby introducing another possible $3 variation in the operating cost.

Experience has shown that the average *on* time for a properly sized gas conversion burner installation will approximate 33⅓ percent. The test data for 33⅓ percent *on* time show a gain of 1.4 percent due to the use of automatically closing air shutters. Since the efficiency at this point was 69½ percent with the shutters open, this 1.4 percent efficiency saving becomes $\frac{1.4 \times 100}{69.5} = 2$ percent, or $2.

These normal variables, which are beyond control and which affect seasonal operating costs, are much greater in magnitude than the possible saving due

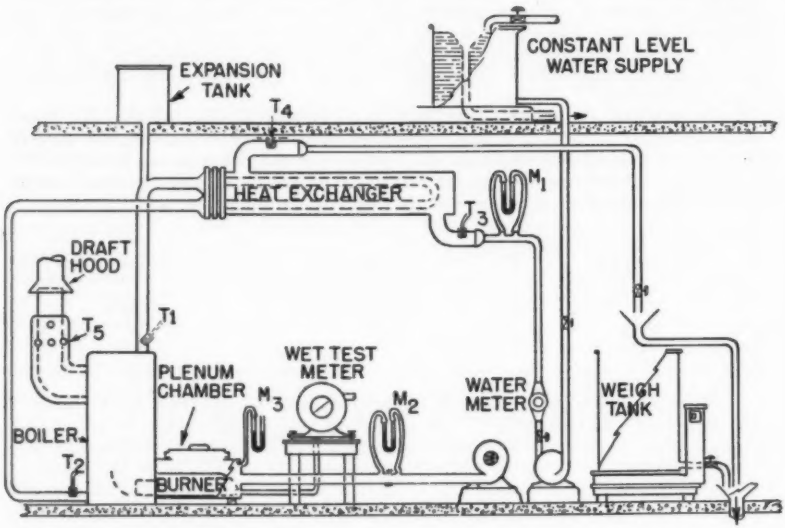

FIG. 1. GENERAL ARRANGEMENT OF EQUIPMENT

to control of *off-period* air. It is concluded, therefore, that it cannot be proved on an actual installation that such control has any value to a customer.

Since it cannot be proved to a customer in his own home that the use of an automatic shutter will save him money, the logical conclusion is that it should be omitted, thus saving both the cost of the development and manufacture. Also, it can be pointed out to the customer that, since a mechanical gadget has been eliminated from the appliance, one possible source of operating trouble and service expense has been eliminated.

The conclusions of this report are directly applicable only to conversion burners of the type used on these tests. These burners are of light weight construction, eliminating the large mass of cast iron or ceramic material characteristic of some of the conversion burners. On a continuous run, a burner having either a low mass or a high mass should give the same thermal efficiency. It is possible that a burner of heavy mass may show a greater loss due to lack of control of *off-period* air.

TEST APPARATUS

Fig. 1 shows the general arrangement of the equipment used in this investigation. The *inshot*[1] type conversion burner was installed in the wet-base boiler. The heated water from the boiler circulates by gravity through the heat exchanger. The heat exchanger is cooled by water from a constant head supply tank located on the floor above. A small pump forces this cooling water through the heat exchanger at a constant rate. The flow of this water is measured by a displacement meter.

By means of the weigh tank, the accuracy of the water meter was established at the beginning and the end of each test. For instantaneous control of the rate of flow, the mercury manometer M_1 was installed across a thin plate orifice in the water line.

The thermocouples at T_3 and T_4 installed in wells at the inlet and outlet connection of the heat exchanger measured the rise in temperature of the cooling water. The thermocouples at T_1 and T_2 indicated the temperature of the circulating hot water at the boiler outlet and the boiler inlet, respectively. An expansion tank was connected to the boiler water system and was located on the floor above the laboratory.

For increased accuracy in measurement of the relatively low temperature, thermocouple installations consisted of groups of seven copper-constantan couples connected in series. Each group was individually calibrated before use.

Fuel gas was measured by the wet test meter, and temperature and pressure were recorded to permit correction of the volume to standard conditions.

The flue gas temperature, T_5, was measured by two thermocouples placed in the flue immediately before the inlet to the drafthood. The smoke pipe was insulated with 2 in. of 85 percent magnesia from the boiler to the drafthood.

To furnish a continuous record of the temperature indicated by the thermocouple installations, a 12-point temperature recorder was employed to give a continuous record of all temperatures. This also furnished a means of determining average temperature during intermittent operation.

A plenum chamber was built to house that part of the gas burner outside of the boiler, so that the flow of air into the boiler during *off-period* could be determined. This chamber was connected to a fan by a duct in which was installed a thin-plate orifice. The flow of air was so adjusted during *off-periods* as to maintain atmospheric pressure in the plenum chamber.

FIRST SERIES OF TESTS

Test Procedure: All tests were made at a rate of heat input slightly less than 140,000 Btu per hr, the maximum input rating of the boiler. The neutral zone was kept at the same position in the furnace for all tests. Although the adjustment of the air shutters could be made to compensate for a change in position of the neutral zone, it was thought best to eliminate this possible variant. At the start of each test, after adjusting the rate of gas input to the burner, the excess air was adjusted to the preassigned value by adjustment of the air

[1] Designates a burner from which the mixture of gas and air leaves the burner in a horizontal direction.

TABLE 1. SUMMARY DATA FOR CONTINUOUS AND INTERMITTENT TESTS—FIRST SERIES (1945)

	Test No.	7	8	6	11	12	14	13
		October						
		4th	4th	3rd	10th	16th	18th	17th
		Continuous					15-Minute Cycle	
1.	Excess Air, percent	30	30	60	60	68	30	30
2.	Time "on", percent	100	100	100	100	100	80	80
3.	Air shutter position, *off* period	....	....	....	....	....	Open	Closed
4.	Number of cycles	....	....	....	....	....	12	12
5.	Total time of test, hours	3.0	3.0	3.0	3.0	3.0	3.0	3.0
	Cooling Water							
6.	Outlet temperature, F deg	160.6	160.3	157.4	153.0	148.2	131.9	132.7
7.	Inlet temperature, F deg	68.8	69.2	70.7	65.9	60.4	61.1	61.8
8.	Temperature difference, F deg	91.8	91.1	86.7	87.1	87.8	70.8	70.9
9.	Volume water metered, cu ft	50.63	50.77	50.38	51.29	50.73	49.89	50.29
10.	Weight of water, corrected, lb	3327	3336	3341	3322	3298	3273	3302
11.	Heat output, Btu	305,365	303,978	289,702	289,346	289,582	231,661	234,153
	Gas							
12.	Volume, metered, cu ft	407.2	408.7	408.8	413.4	418.2	326.9	326.3
13.	Reduction factor	0.965	0.960	0.968	0.951	0.950	0.949	0.950
14.	Volume, corrected to S.C. cu ft	392.9	392.2	395.9	393.3	397.3	310.1	309.9
15.	Heating value, corrected, Btu/cu ft	1045	1045	1054	1049	1051	1050	1054
16.	Total heat input, Btu	410,591	409,880	417,434	412,547	417,583	325,637	326,601
	Flue Loss, *on*							
17.	Average flue-gas temp, F deg	538	538	575	575	577	515	525
18.	Room temperature, F deg	73.7	74.6	72.9	75.4	74.6	75.7	75.1
19.	Temperature difference, F deg	464.3	463.4	502.1	499.6	502.4	439.3	449.9
20.	Average O_2 content, percent	4.8	4.7	8.4	8.2	8.8	4.6	4.9
21.	Flue loss, from chart, percent	21.4	21.3	25.4	25.1	25.8	20.6	21.0
	Air Loss, *Off*							
22.	Average flue temperature, F deg						282	
23.	Room temperature, F deg						75.7	
24.	Temperature difference, F deg						206.3	
25.	Total *off-period* time, hours						0.43	
26.	Avg differential across orifice, in. H_2O						1.117	
27.	Avg air flow into boiler, lb/hr						92.5	
28.	Loss for total *off* time, Btu						1951	
29.	Off-period air loss, percent						0.6	
	Summary, Heat Balance							
30.	Boiler output, percent	74.4	74.2	69.4	70.1	69.3	71.1	71.7
31.	Flue Loss, *on* period, percent	21.4	21.3	25.4	25.1	25.8	20.6	21.0
32.	Air loss, *off* period, percent	....	....	....	....	....	0.6	....
33.	Radiation, etc. losses, percent	4.2	4.5	5.2	4.8	4.9	7.7	7.3

shutters. The rate of flow of cooling water was adjusted and maintained at a selected value. The average of the calibration constants of the water meter obtained before and after the test was used in calculating the rate of flow of cooling water.

For the continuous tests, a warm-up period of approximately two hours was required before equilibrium conditions were obtained. The continuous record of temperatures furnished by the temperature recorder gave visual indication when this was accomplished.

Each test was started and stopped *on the run*. As all readings of the flow of gas, water and temperatures could not be made simultaneously, a time schedule was followed and each item was recorded at definite intervals. Continuous samples of the flue products were obtained and analyzed at half-hour intervals.

Intermittent tests were run in much the same way as the continuous tests. A special timer was used which could be set to give a time cycle of the desired length, and which permitted variations in the percentage *on* time to be made as required.

TABLE 1. SUMMARY DATA FOR CONTINUOUS AND INTERMITTENT TESTS—FIRST SERIES (1945)—*Continued*

24	23	18	17	19	16	15	21	20	22
NOVEMBER		OCTOBER		NOVEMBER	OCTOBER			NOVEMBER	
29th	26th	26th	25th	2nd	22nd	19th	12th	9th	24th
15-MINUTE CYCLE					30-MINUTE CYCLE				
30	30	30	30	30	30	30	30	30	30
50	50	20	20	20	80	80	20	20	20
Open	Closed	Open	Closed	Closed	Open	Closed	Open	Closed	Closed
12	20	15	12	11	7	4	4	3	10
3.13	5.05	3.89	3.02	2.79	3.5	2.0	2.02	1.51	5.04
94.3	95.2	80.9	77.0	80.4	135.2	135.8	77.0	80.8	69.7
50.9	49.6	62.1	58.5	61.6	61.8	63.5	57.4	61.3	51.1
43.4	45.6	18.8	18.5	18.8	73.4	72.3	19.6	19.5	18.6
46.53	74.62	63.95	49.56	46.45	57.72	33.63	31.66	24.50	75.11
5440	5577	4340	3327	3142	3851	2223	2196	1673	5427
169,210	254,255	81,597	61,618	58,979	282,917	160,590	43,005	32,693	101,004
214.9	345.1	121.1	93.6	89.56	396.9	227.5	64.83	48.55	147.2
0.954	0.983	0.951	0.949	0.934	0.950	0.940	0.965	0.949	0.974
205.1	339.3	115.2	87.9	83.7	377.0	213.9	62.54	46.09	143.3
1041	1043	1054	1054	1024	1035	1045	1037	1034	1048
213,520	353,921	121,365	92,657	85,699	390,191	223,484	64,854	47,657	150,220
683	482	443	425	420	533	524	455	470	435
74.9	72.3	77.8	75	78.7	74	78	77	77	67
608.1	409.7	365.2	350	341.6	459	446	378	393	368
4.7	4.8	4.8	4.8	6.0	4.6	4.8	4.7	5.1	5.0
19.9	20.0	18.9	18.4	18.8	21.1	20.9	19.1	19.7	19.0
209.7	222	153.4		156.3	251		137	141	134
74.9	72.3	77.8		78.7	74.4		77	77	67
134.8	149.7	75.6		77.6	176.6		60	64	67
1.59	2.49	3.08		2.2	0.64		1.59	1.20	4.0
2.065	0.384	0.481		0.116	0.832		1.76	0.383	0.229
130.7	56.6	63.8		30.3	81.8		119.2	56.0	43.7
6067	5023	3536		1231	2200		2713	1025	2803
3.1	1.4	2.9		1.4	0.6		4.2	2.2	1.90
69.9	71.8	67.2	66.5	68.8	72.5	71.9	66.3	68.6	67.2
19.9	20.0	18.9	18.4	18.8	21.1	20.9	19.1	19.7	19.0
3.1	1.4	2.9	....	1.4	0.6	....	4.2	2.2	1.9
7.1	6.8	11.0	15.1	11.0	5.8	7.2	10.4	9.5	11.9

For the intermittent tests, the meter readings for the gas and cooling water were made at the beginning of each cycle. Samples of the flue gas were taken only during the *on* period.

During the period of each test, the heating value of the gas was determined in a Junkers-type gas calorimeter. These determinations were carefully made and checked, as a change in only a few Btu per cubic foot is directly reflected in the percentage values of the resulting heat balance.

The temperature of the water leaving and returning to the boiler was of the order of 190 F and 160 F in the continuous tests. Thus, the conditions closely approximated those of a normal residential installation.

DISCUSSION OF RESULTS OF FIRST SERIES

Table 1 presents the observed and calculated results of the first series of tests made in this investigation. A summary heat balance is shown for each test at the bottom of the column.

TABLE 2. SUMMARY OF HEAT-BALANCE FOR INTERMITTENT TESTS
(30 Percent Excess Air)

	80 PERCENT *on*		50 PERCENT *on*		20 PERCENT *on*		
Percentage of Heat Input—15-min cycle	Shutters Open	Shutters Closed	Shutters Open	Shutters Closed	Shutters Open	Shutters Closed	
Test No.	14	13	24	23	18	17	19
Boiler Efficiency	71.1	71.7	69.9	71.8	67.2	66.5	68.8
Flue Loss *on*	20.6	21.0	19.9	20.0	18.9	18.4	18.8
Flue Loss *off*	0.6	——	3.1	1.4	2.9	——	1.4
Radiation, etc losses by diff	7.7	7.3	7.1	6.8	11.0	15.1	11.0
TOTAL	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Percentage of Heat Input—30-min cycle							
Test No.	16	15			21	20	22
Boiler Efficiency	72.5	71.9			66.3	68.6	67.2
Flue Loss *on*	21.1	20.9			19.1	19.7	19.0
Flue Loss *off*	0.6	——			4.2	2.2	1.9
Radiation by diff	5.8	7.2			10.4	9.5	11.9
TOTAL	100.0	100.0			100.0	100.0	100.0

At the start of the investigation, a schedule of 50 tests was prepared, which systematically considered the effect of progressive changes in each of the variable factors in relation to changes of one or more of the other. It was found, however, that the essential information was given by a considerably smaller number of tests and the schedule was accordingly curtailed.

A heat balance, in which the heat in the flue gas is divided between the sensible heat in the dry gas and the heat in the steam, could not be calculated, as an analysis of the natural gas was not made. Furthermore, the flue-gas temperatures obtained may not be exact. The primary purpose of the couples as installed was to obtain temperatures for comparison and control. Shielded or possibly high-velocity thermocouples might be necessary for exact measurements.

Of the several items involved in the heat balance, the total heat input and the boiler output were determined directly. When determined, the *off-period* air loss likewise was found directly from observed data. The sensible heat loss of the wet products of combustion leaving the boiler was determined by use of a chart of flue losses for a natural gas of *normal* composition. It involved the temperature and oxygen content of the flue gas. The flue losses determined in this way, although indirect and possibly not precise, are believed to give results of sufficient accuracy for purposes of relative comparison between tests. The radiation and unaccounted-for losses were determined by difference.

Continuous Tests: Five continuous tests were made, each of three hours, duration. Of these, Tests Nos. 7 and 8 were made with 30 percent excess air, Tests Nos. 6 and 11 with 60 percent excess air, and Test No. 12 with 68 percent

excess air, which was the maximum obtainable with the shutters fully opened. The data and calculated results are shown in Table 1 under these test numbers.

The effect of changing the percentage of excess air on the results of the continuous tests is shown in the summary heat-balance results in Table 1. As was to be expected, the boiler efficiency values decreased as the percentage of excess air was increased, and the flue losses increased.

Intermittent Tests: Table 2 summarizes the heat balances of the series of intermittent tests in which the effect of leaving the air shutters open or closed in the *off-period* was investigated for cycles of 15 and 30 min and for varying percentages of time *on* in each cycle.

Effect of Open or Closed Shutters: With the exception of the tests at 80 percent *on* in the 30-min cycle where the efficiency was 0.6 percent higher with the shutters open than closed, and in one of the repeat tests with the shutters closed at 20 percent *on* in the 15-min cycle where the efficiency was 0.7 percent less than with the shutters open, the efficiency was definitely, though only slightly, higher when the shutters were closed in the *off-period* than when they were open.

The loss in sensible heat in the air passing through the boiler in the *off-period* was not measured in all tests with the shutters closed, but, in those pairs of tests where it was measured, the difference in the loss was nearly the same as the difference in efficiency.

The reason for the small difference in the loss in the air flowing through the boiler in the *off-period*, whether the shutters were open or closed, can be seen by reference to the data in Table 1. Although the measured average rate of air flow through the boiler with the shutters open was about twice the flow with the shutters closed, the temperature of this air in the flue was so low for either condition that the total heat loss was small. Hence, the differences could not be large.

EFFECT OF DESIGN OF BURNER

Because the mass of material in the burner used in these tests was small, there was little heat stored in the burner at the end of an *on* period. With a different design of burner, as, for example, one with refractory radiants which are heated to a high temperature, it can be reasoned that the effect of open or closed shutters would be greater.

Certainly the air entering the boiler in the *off-period* would be more highly heated. Because of the higher temperature, the rate of flow in the *off-period* would be higher than with the burner used in these tests, both with the shutters closed and open. But, if the rate of air flow through the unit with the shutters open were again twice that with them closed, and the loss were greater even with the shutters closed for the burner of larger mass, the net effect of closed or open shutters would be greater.

SECOND SERIES OF TESTS

Twenty-four complete efficiency tests were run in this series.

Test Apparatus and Procedure: The same general arrangement of the apparatus and equipment was used as in the earlier tests. Several minor changes in instrumentation were made in an effort to obtain more exact and reliable data.

TABLE 3. SUMMARY DATA FOR "a" AND "b" TESTS (1946)

	Test No.	3a	9a	10a	1b	4a	15a	16a	17a
		April 26	May		July 15	April 29	June		July
			28	29			25	27	
		15-Minute Cycle							
1.	Excess air, percent	30	30	30	30	30	30	30	30
2.	Time *on*, percent	20	20	20	20	20	20	20	20
3.	Air shutter position, *off* period	Closed	Closed	Closed	Closed	Open	Open	Open	Open
4.	Number of cycles	18	18	14	14	17	18	16	16
5.	Total time of test, hours	4.6	4.6	3.5	3.6	4.3	4.6	4.0	4.1
	Cooling Water								
6.	Outlet temperature, F deg	8.49	88.8	86.8	103.8	81.5	96.7	98.3	100.6
7.	Inlet temperature, F deg	65.7	68.5	67.2	83.2	62.9	76.9	79.0	81.4
8.	Temperature difference, F deg	19.2	20.3	19.6	20.6	18.6	19.8	19.3	19.2
9.	Volume water metered, cu ft	76.9	78.5	60.4	67.1	72.2	79.7	70.1	72.4
10.	Weight of water, corrected, lb	5042	5112	3925	4383	4725	5040	4453	4603
11.	Heat output, Btu	96,797	103,778	76,893	90,340	87,980	100,094	86,084	88,55
	Gas								
12.	Volume, metered, cu ft	144.0	148.4	108.8	135.6	132.1	154.4	131.7	135.3
13.	Reduction factor	0.929	0.949	0.954	0.936	0.964	0.921	0.926	0.93
14.	Volume, corrected to S.C., cu ft	133.9	140.8	103.8	126.9	127.4	142.2	122.0	126.8
15.	Heating value, corrected, Btu/cu ft	1015	1034	1028	999	1037	1013	996	995
16.	Total heat input, Btu	135,753	145,448	106,644	126,677	131,972	143,924	121,386	126,08
	Flue Loss, *On*								
17.	Average flue-gas temp, F deg	446	448	444	465	441	458	456	457
18.	Room temperature, F deg	77.6	76.5	75.6	83.3	75.4	86.7	86.1	83.2
19.	Temperature difference, F deg	368.4	371.5	368.4	381.7	365.6	371.3	369.9	373.8
20	Average O_2 content, percent	5.7	5.3	5.1	4.7	5.5	4.7	5.0	5.0
21	Flue loss, from chart, percent	19.4	19.2	19.1	19.4	19.2	19.4	19.5	19.7
	Air Loss, *Off*								
22.	Average flue temperature, F deg	177	178	177	201	170	187	184	208
23.	Room temperature, F deg	77.6	76.5	75.6	83.3	75.4	86.7	86.1	83.2
24.	Temperature difference, F deg	99.4	101.5	101.4	117.7	94.6	100.3	97.9	124.8
25.	Total *off-period* time, hours	3.62	3.59	2.79	3.22	3.39	3.58	3.16	3.24
26.	Average differential across orifice, in. H_2O	0.341	0.335	0.319	0.350	2.16	1.98	1.91	2.01
27.	Average air flow into boiler, lb/hr	28.6	29.0	28.0	29.1	72.2	68.0	67.6	69.1
28.	Loss for total *off* time, Btu	2450	2520	1890	2614	5530	5800	4920	6650
29.	Off-period air loss, percent	1.8	1.7	1.8	2.0	4.2	4.0	4.0	5.3
	Summary, Heat Balance				Avg				
30.	Boiler output, percent	71.3	71.4	72.1	71.3 71.5	66.7	69.5	70.9	70.2
31.	Flue loss, *on* period, percent	19.4	19.2	19.1	19.4 19.3	19.2	19.4	19.5	19.7
32.	Air loss, *off* period, percent	1.8	1.7	1.8	2.0 1.8	4.2	4.0	4.0	5.3
33.	Radiation, etc., losses, percent	7.5	7.7	7.0	7.3 7.4	9.9	7.1	5.6	4.8

For increased accuracy in measurement of the relatively small volume of air flowing through the boiler during the *off-period,* precision-type inclined micromanometers of improved accuracy in reading and manipulation were substituted for those used during the original series of tests. In addition, a more sensitive zero gage for indicating maintenance of atmospheric pressure conditions within the plenum chamber was substituted for the gage formerly used.

To give easier and more positive draft control, and to offset the effect of fluctuations of draft in the breeching duct, because of the load placed on it by other laboratory installations made since the original series of tests, a butterfly damper and barometric damper were inserted between the drafthood and breeching. A precision-type inclined micromanometer was utilized to determine the magnitude of draft conditions within the stack.

To minimize the possibilities of error in measurement of inlet and outlet cooling water temperatures, and thereby to give more accurate determinations of thermal output, a high sensitivity type portable precision potentiometer was used instead of the 12-point potentiometer recorder employed for the original tests.

TABLE 3. SUMMARY DATA FOR "a" AND "b" TESTS (1946)—*Continued*

16a	17a	8a	2b	7a	3b	1a	4b	7b	2a
	JULY 3	MAY 20	JULY 16	MAY 17	JULY 17	APRIL 24	JULY 18	AUGUST 9	APRIL 25
27									
				15-MINUTE CYCLE					
30	30	30	30	30	30	30	30	30	30
20	20	33⅓	33⅓	33⅓	33⅓	50	50	50	50
Open	Open	Closed	Closed	Open	Open	Closed	Closed	Closed	Open
16	16	16	16	16	16	17	14	12	16
4.0	4.1	4.0	4.0	4.0	4.0	4.3	3.5	3.0	4.1
98.3	100.6	98.8	113.7	99.2	113.1	110.7	129.5	123.2	108.3
79.0	81.4	68.7	82.5	68.6	81.5	66.4	82.8	80.3	64.9
19.3	19.2	30.1	31.2	30.6	31.6	44.3	46.7	42.9	43.4
70.1	72.4	66.7	67.7	67.3	69.8	72.8	57.9	51.9	69.22
53	6693	4389	4352	4430	4425	4682	3680	3276	4505
86,084	98,554	132,065	135,873	135,424	139,880	207,418	171,667	140,556	195,526
31.7	131.3	197.2	204.3	201.9	212.1	311.0	268.8	227.8	294.5
0.926	0.937	0.928	0.934	0.942	0.925	0.935	0.916	0.911	0.928
22.0	126.8	183.1	190.9	190.1	196.2	290.8	246.2	207.5	273.1
96	991	1023	1000	1039	1014	1040	1026	1004	1021
21,386	126,087	187,009	190,900	197,445	198,896	302,384	252,520	208,245	278,876
56	457	472	502	486	502	515	502	514	502
86.1	83.2	81.3	83.5	77.1	86.1	78.6	88.4	87.1	79.2
69.9	373.8	390.7	418.5	408.9	415.9	436.4	413.6	426.9	422.8
5.0	5.0	4.9	5.5	5.5	4.6	5.5	5.0	5.3	5.7
19.5	19.7	19.8	20.5	20.5	20.0	21.0	20.2	20.6	20.9
84	208	213	234	207	222	257	249	266	231
86.1	83.2	81.3	83.5	77.1	86.1	78.6	88.4	87.1	79.2
97.9	124.8	131.7	150.5	129.9	135.9	178.4	160.6	178.9	151.8
3.16	3.24	2.62	2.58	2.62	2.62	2.09	1.70	1.50	2.01
1.91	2.01	0.390	0.423	2.24	2.22	0.406	0.465	0.50	2.84
67.6	69.1	30.8	32.3	73.6	72.0	31.3	33.2	34.0	81.5
20	6650	2528	2991	5950	6048	2771	2154	2172	5915
4.0	5.3	1.4	1.6	3.0	3.0	1.0	0.9	1.0	2.1
		Avg	Avg		Avg				Avg
70.2	70.2	69.3 70.6	71.2 70.9	68.6	70.4 69.5	68.6	68.0	67.5	68.0 70.1
19.5	19.7	19.5 19.8	20.5 20.1	20.5	20.0 20.2	21.0	20.2	20.6	20.6 20.9
4.0	5.3	4.4 1.4	1.6 1.5	3.0	3.0 3.0	1.0	0.9	1.0	1.0 2.1
5.6	4.8	6.8 8.2	6.7 7.5	7.9	6.6 7.3	9.4	10.9	10.9	10.4 6.9

Special precautions were taken to maintain constant the rate of heat input, the percentage of excess air, the rate of flow of cooling water, and the time schedule for all readings.

All tests were run on a 15-min cycle of operation, with 30 percent excess air, constant rates of heat input during *on-periods*, and constant flow of cooling water in the heat exchanger at all times. The percentage of *on* time within the operating cycle and the position of the air shutters during *off-periods*, whether open or closed, were varied as required by the schedule of tests.

Determination of Heat Balance: No changes were made in the original method for determination of the thermal efficiency of the boiler or *off-period* air loss. The semi-calorimetric method previously described was used to determine the boiler efficiency. The actual *off-period* air loss, as sensible heat, was determined by measuring the rate of flow of air and its rise in temperature in passing through the boiler during the *off-period*.

Schedule of Tests: For the determination of the overall boiler efficiency and *off-period* air losses during the performance of the original series of tests

TABLE 3. SUMMARY DATA FOR "a" AND "b" TESTS (1946)—*Continued*

12a	11a	13a	14a	5a	5b	6a	6b
JUNE 10	MAY 31	JUNE 17	JUNE 21	MAY 15	JULY 23	MAY 16	JULY 24
15-MINUTE CYCLE							
30	30	30	30	30	30	30	30
50	50	50	50	66⅔	66⅔	66⅔	66⅔
Open	Open	Open	Open	Closed	Closed	Open	Open
18	18	14	16	12	12*	12	12
4.5	4.5	3.5	4.0	3.0	3.0	3.0	3.0
119.7	115.1	122.6	118.6	124.7	140.3	125.9	142.0
76.0	70.9	76.4	74.2	66.8	80.8	69.3	82.2
43.7	44.2	46.2	44.4	57.9	59.5	56.6	59.8
76.28	75.67	60.12	69.02	50.4	52.3	50.7	52.2
4927	4908	3851	4446	3310	3313	3335	3297
215,205	216,916	178,109	197,638	191,573	197,203	188,745	197,373
327.9	327.3	266.2	298.7	285.0	299.1	291.0	299.8
0.941	0.937	0.932	0.939	0.946	0.931	0.930	0.925
308.5	306.5	248.2	280.6	269.6	278.5	270.5	277.4
1007	1003	1025	992	1016	1001	1003	1009
310,626	307,396	254,324	278,375	273,905	278,759	271,253	279,848
498	533	513	496	535	555	521	531
82.1	78.7	82.3	80.3	77.0	82.6	79.9	85.0
415.9	454.3	430.7	415.7	458.0	472.4	441.1	446.0
5.1	5.1	5.1	5.2	5.6	5.4	5.6	5.2
20.3	20.7	20.6	20.2	22.2	22.0	21.2	21.1
237	231	242	232	306	320	281	283
82.1	78.7	82.3	80.3	77.0	82.6	79.9	85.0
154.9	152.3	159.7	151.7	229.0	237.4	201.1	198.0
2.20	2.21	1.69	1.94	0.974	0.974	0.959	0.953
2.56	2.53	2.84	2.41	0.455	0.540	2.66	2.89
78.0	77.0	81.8	75.4	33.2	36.0	78.9	82.0
6308	6160	5260	5240	1755	1980	3620	3670
2.0	2.0	2.1	1.9	0.6	0.7	1.3	1.3
			AVG		AVG		AVG
69.3	70.6	70.0	71.0 70.2	69.9	70.7 70.3	69.6	70.5 70.0
20.3	20.7	20.6	20.2 20.5	22.2	22.0 22.1	21.2	21.1 21.2
2.0	2.0	2.1	1.9 2.0	0.6	0.7 0.7	1.3	1.3 1.3
8.4	6.7	7.3	6.9 7.3	7.3	6.6 6.9	7.9	7.1 7.5

* Equilibrium conditions maintained for 16 cycles—no data for 5 cycles.

employing cyclic operation, tests had been run at 80, 50 and 20 percent *on-period* time. Duplicate tests had been run in only two instances.

In order to establish more definitely the effect of decreasing percentages of *on-period* time, additional tests at 33⅓ and 66⅔ percent *on* time also were planned for the second series of tests. Because a gas burner is rarely operated on an 80 percent *on* time schedule, no additional tests of this percentage were scheduled. The tests originally planned for the second series were single tests of 66⅔, 50, 33⅓ and 20 percent *on* time within 15-min schedules, with the air shutters open, and a similar series with them closed.

Several inconsistencies were found in the results for these tests, and it was decided to make a series of duplicate tests in an effort to establish the reproducibility of the calculated results. Some of the tests were repeated a number of times.

DISCUSSION OF RESULTS OF SECOND SERIES

Table 3 presents a summary of the observed and calculated data obtained in the second series of tests. For each test the data are divided horizontally

into six sections, each containing data relating to miscellaneous items, cooling water, gas, flue loss, air loss, and a summary heat balance.

Effect of Open or Closed Shutters: Fig. 2 is a graphical representation of data from Table 3. The changes in boiler efficiency, flue loss, radiation loss and *off-period* air loss, expressed as percentage of total heat input, are shown in relation to percentages of *on* time. The plot was constructed using average

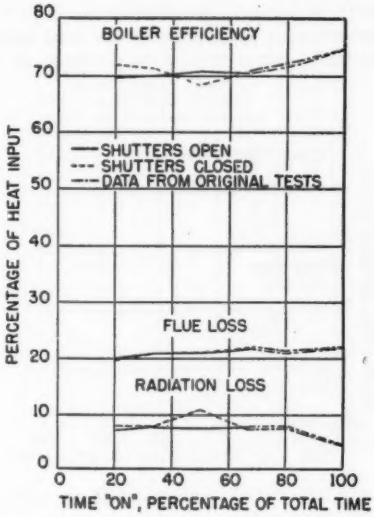

FIG. 2. EFFECT OF VARYING THE PERCENTAGE *on* TIME IN A 15-MIN CYCLE. BASED ON PERCENTAGE OF TOTAL HEAT INPUT

values for each series of tests. Data from the first series of tests were used for 80 and 100 percent *on* time.

For tests performed with the shutters open, the increase in boiler efficiency with increase in percentage *on* time is small, but fairly regular. For the tests with the shutters closed, however, the average value for the tests at 50 percent *on* time is obviously out of line, as compared with the values for other series of tests. To check this value, three complete tests were made, with extreme care taken in flow measurement, time regulation, and recording of data. For these tests, the deviation of the thermal efficiency from the average was ± 0.5 percent.

As the average percentage radiation loss for these same tests is out of line, practically to the same extent as the boiler efficiency, and since the radiation losses were obtained by difference, it must be concluded that, for some unexplained reason, the boiler efficiencies are several percent too low. A careful

examination of the data furnished no explanation for the deviation shown in the plot. The tests were not run consecutively.

If the value for the 50 percent *on* tests, shutters closed, is disregarded, the plots for percentage boiler efficiencies, Fig. 2, show the following trends: (1) the boiler efficiency increases slightly as the percentage *on* time is increased, (2) the increase is somewhat less than indicated by results of the single determinations shown in Table 1, and (3) the percentage boiler efficiencies for those tests with the shutters closed are slightly higher than comparable tests with the shutters open, but the differences are relatively small.

The flue loss, expressed as a percentage of the total heat input, is shown in Fig. 2 to increase gradually, but to a very small extent, as the percentage of *on* time is increased.

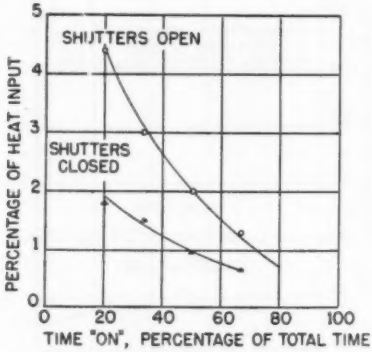

FIG. 3. EFFECT ON *off-period* AIR LOSS WHEN VARYING THE PERCENTAGE *on* TIME IN A 15-MIN CYCLE. BASED ON PERCENTAGE OF TOTAL HEAT INPUT

Disregarding the value at 50 percent *on* time, with the air shutters closed, the radiation losses shown in the plot remain fairly constant over the range of percentage time *on* from 20 to 80 percent.

Fig. 3 shows the *off-period* air loss, which decreases regularly as the percentage of *on* time increases. It is evident that the direct determination of the *off-period* air loss by measuring the flow of air and its temperature in passing through the boiler, furnishes a reliable method for evaluation of the magnitude of the air loss, and for evaluation of the loss of heat energy occasioned by not closing the air shutters during *off-period*.

The air losses determined by the direct method with the shutters open are about twice those with the shutters closed. The losses, however, are small for either condition. The results confirm the conclusion reached in the first series of tests.

Table 4 shows a comparison of the magnitude of differences in percentage values for the *off-period* air loss for open and closed shutters, as determined by the direct method and by the difference in calculated boiler efficiencies.

The differences are shown for several percentages of *on* time. Because of the seeming abnormality of the determinations of boiler efficiency for 50 percent *on* time operation with the shutters closed, no comparison is possible. However, for all other percentages of *on* time, the agreement between the two methods is good.

The directly determined difference in sensible heat loss between that with shutters open and that with shutters closed remains rather constant over the range of percentage *on* times when referred to the rated input of the heater. For example, at 20 percent *on* time, the additional rate of loss is $2.6 \times 0.20 = 0.52$

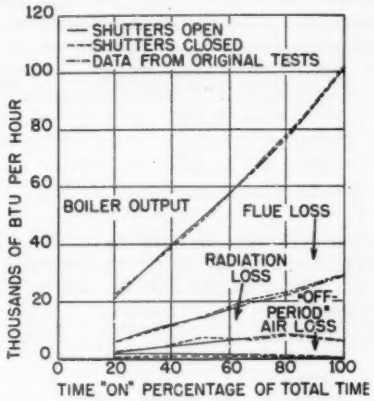

FIG. 4. EFFECT OF VARYING THE PERCENTAGE *on* TIME IN A 15-MIN CYCLE. BASED ON ACTUAL VALUES OF BTU PER HOUR INPUT

percent of rated input; at 66⅔ percent *on* time, the additional rate of loss is $0.6 \times 0.66 = 0.40$ percent of rated input. Thus, over a range of partial loads, the savings in heat from the use of shutters are nearly constant and are small.

Fig. 4 shows a plot of the change in boiler output, flue loss, radiation loss, and *off-period* air loss, in terms of Btu per hour, with changing percentages of *on* time.

POSSIBLE ERRORS

Early in the investigation, it was recognized that quite accurate measurements of the several factors involved in the tests would be necessary. It was further recognized that cyclic operation would complicate the determination of mean values for these factors, and that small errors would be possible in the measurement of rate of flow, temperature, and time. Whether these errors would prove to be compensating or compounding would affect the numerical value for the boiler efficiencies obtained.

It was hoped that the results for tests under identical conditions could be duplicated within a maximum range of ±0.5 percent. The results of these tests show that a more likely maximum range of reproducibility is of the order of ±1.0 percent. For the tests performed in this investigation, the range of differences in boiler efficiencies for identical tests, expressed as maximum deviations from the average value, are shown in Table 5.

Owing to the fact that differences between *off-period* air losses for open and closed shutters appear to be of the same order as the accuracy of the tests,

TABLE 4. COMPARISON OF THE MAGNITUDE OF DIFFERENCES IN PERCENTAGE OF *Off-Period* AIR LOSS FOR OPEN AND CLOSED AIR SHUTTERS, AS DETERMINED BY TWO METHODS AT SEVERAL PERCENTAGES OF *On Time*

ON TIME, PERCENT	DIFFERENCE BY DIRECT DETERMINATION, PERCENT	BY DIFFERENCE IN BOILER EFFICIENCY, PERCENT
20	2.6	2.2
33⅓	1.5	1.4
50	1.0	—
66⅔	0.6	0.3

TABLE 5. MAXIMUM DEVIATIONS OF CALCULATED PERCENTAGES OF BOILER EFFICIENCY, BASED ON AVERAGES FOR IDENTICAL TESTS

TIME *ON*, PERCENT	POSITION OF SHUTTERS	NUMBER OF TESTS	DEVIATION FROM AVERAGE, PERCENTAGE OF HEAT INPUT
20	Closed	4	±0.6
20	Open	4	±2.6[a]
33⅓	Closed	2	±0.3
33⅓	Open	2	±0.9
50	Closed	3	±0.5
50	Open	5	±0.9
66⅔	Closed	2	±0.4
66⅔	Open	2	±0.5

[a] If Test No. 4a is omitted, the deviation becomes ±0.7 percent.

comparison based on single tests cannot be made. However, averages of a number of tests do give a fairly reliable indication of small differences, as the comparison of data shown in Table 4 indicates.

The small errors in the measurement of temperature differentials become increasingly significant as the temperature difference becomes smaller. To insure greater precision than was afforded by use of the temperature recorder, the temperatures of the inlet and outlet cooling water in the heat exchanger were determind by use of a manual precision potentiometer indicator. This procedure was used for tests beginning with number 9*a*.

To check reproducibility of results, a series of 4 tests was made at 20 percent *on* time and another series of 5 tests at 50 percent *on* time, all with air shutters open during *off-periods*. For these tests, results were computed on an hourly basis as well as for the total elapsed time. It was found that the agreement between these results calculated on an hourly basis was usually no better than that previously found for the agreement between individual tests.

Two other possible sources of error were investigated, the automatic timer used for the cyclic operation and the water meter used to measure the flow of cooling water to the heat exchanger. The timer was found to be occasionally in error in timing of the *on* and *off* periods, although the error in each cycle was on the order of only 5 to 20 seconds.

During the last two tests (*6b* and *7b*), in addition to manually recording the temperature readings of the water entering and leaving the heat exchanger, the water flowing through the heat exchanger was weighed during alternate cycles, and the timer was operated manually. There was some improvement in the general agreement between hourly calculated efficiencies, but the efficiency values varied by as much as 0.8 percent.

The calorific value of the natural gas showed considerable variation from day to day, particularly in the spring, early summer, and fall. Throughout the second series of tests, the lowest calorific value, 992 Btu per cu ft, was found on June 21, and the highest value, 1040 Btu per cu ft, was found on June 24.

Because a change of only 10 Btu in the calorific value of the gas changes the calculated over-all boiler efficiency by about 0.6 percent, it was necessary to determine carefully the calorific value during the progress of each test. Duplicate determinations were made at least twice during the course of each test. It is not likely that the calorific value of the gas would change appreciably during the course of a 3- or 4-hr test. Changes usually occurred over a period of days, rather than hours. There has been some indications, however, that the calorific value has changed slightly during a test.

It is probable that the differences in results shown by some tests were owing to the compounding instead of the compensation of errors, rather than from the inaccuracy of a single factor.

ACKNOWLEDGMENT

This paper is a report of data obtained and conclusions reached in a series of tests sponsored by the Surface Combustion Corp. at Battelle Memorial Institute, Columbus, Ohio.

DISCUSSION

CARL H. FLINK, New York, N. Y.: All gas appliances are equipped with a draft hood at the outlet of the appliance. This draft hood is so constructed that, if the outlet of that draft hood is blocked, the appliance will still operate because other openings permit the products of combustion to escape into the space surrounding the boiler. If a gas appliance were made air-tight at the bottom and this air shutter described by Mr. Myler were able to close off the air completely at the bottom of that appliance, obviously no air could enter at the bottom of the boiler or furnace when the gas valve is closed. However, since the boiler or furnace is warmer than the air of the space surrounding the appliance, it is obvious that any air in the flues which is warmer than the basement would pass up through the boiler and escape into the basement, while cold air would enter through the draft hood to replace it. That would act, I think, to further reduce any saving that might be made by the use of air shutters at the bottom of the appliance.

R. K. THULMAN, Washington, D. C.: I would like to ask if the effectiveness of the air shutter might not be increased in the case of a boiler other than of the wet base type; in other words, with a large mass of material at a higher temperature

at the inner surface of the boiler; and second, what effect there might be under different kinds of temperatures, particularly with reference to the mass involved in the temperature.

W. J. WATCHLER, Cleveland, Ohio: Why was the wet-base boiler and an inshot burner chosen at the place where the work was done?

AUTHORS' CLOSURE: Mr. Flink's discussion raised a question which is covered in the written paper and which I should have stressed in this presentation. In the design of this automatic air shutter, it is not the intention to design a shutter mechanism which would give absolute air closure. Such a thing is neither practicable nor desirable. Air for combustion must be provided for the constant burning pilot. The data show that the off-period air flow through the boiler with the air shutter closed is about half that with the air shutter opened. I believe this is fairly typical of the effect of an automatic air shutter on most conversion burners that have been so equipped.

Replying to Mr. Thulman, we did not feel that the choice of the type of boiler particularly entered into the results. Whether the boiler was a wet-base or a dry-base would not have made any particular difference in the temperature of the material inside the boiler. The design of the burner, as to whether it was a light weight sheet metal or a heavy cast iron one with a refractory or other type radiance, may have an effect. As we pointed out, these results apply only to a light weight type of burner which is the type most commonly built today.

Replying to Mr. Watchler, we used a wet-base boiler because it was representative of many new installations today and because one was conveniently available. The inshot burner was used because it was designed specifically for wet-base boilers.

1356

No. 1356

PERFORMANCE OF FOURTEEN MASONRY CHIMNEYS UNDER STEADY STATE CONDITIONS

By PAUL R. ACHENBACH* AND SELDEN D. COLE,** WASHINGTON, D.C.

INTRODUCTION

A CHIMNEY has two characteristics of prime importance, aside from considerations of cost and appearance, namely, the magnitude of the draft it will produce and its ability to convey gases at high temperature without igniting combustible materials in the structure of which it is a part. The *National Board of Fire Underwriters* has accepted chimneys made of solid brick, concrete, solid-molded or solid-cast concrete chimney units, or burned clay, during the past several years for warm-air, hot-water, or low-pressure steam heating systems. Walls 8 in. thick are required if the chimney is unlined, and walls 3¾ in. thick may be used if a suitable lining is provided. The drafts produced by lined brick chimneys of the cross-sectional areas and heights usually found in domestic use were determined at the *National Bureau of Standards* and are now published in THE GUIDE of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. However, the construction of small dwellings on a mass production basis in recent years has created a demand for chimney materials that would be cheaper, more easily erected, lighter in weight, or for some other reason more advantageous to use than brick.

When construction materials other than brick are used for a chimney, the question naturally arises whether such a chimney is the equivalent of a brick chimney as a draft producer and in fire-resistance properties. In order to provide information on various types of chimney materials, 14 chimneys of equal height were constructed at the *National Bureau of Standards,* using brick, shale tile, and cinder concrete blocks with fire clay and metal liners.

OBJECT OF INVESTIGATION

The objectives of the investigation were as follows:

1. Measurement of the draft produced by chimneys made of the various materials under steady state conditions for a range of mass flows of the flue gases and for a range of temperatures.

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Presented at the 55th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1949.

2. Comparison of the draft produced by metal-lined chimneys and refractory-lined chimneys.

3. Comparison of the performance of round and rectangular flue liners of equal cross-sectional area and of round and square flue liners of equal principal dimensions.

4. Measurement of the effect on the draft produced when one of two contiguous liners in the same chimney carried hot gases while the other was open.

5. Observation of the effect on the draft produced when the base of a chimney was sealed at the bottom of the thimble.

6. Determination of the effect of grouting in the liner with mortar on the available draft.

The fire-resistance properties of the same 14 chimneys and some additional specimens have been studied by the *Fire Protection Section* of the *National Bureau of Standards* and results on this phase of the investigation will be published by that Section in the near future. The performance of selected specimens

TABLE 1. COMBUSTIBLE EQUIVALENTS OF FLUE-GAS FLOW RATES

FUEL OIL RATE GAL/HR	CO_2 IN FLUE GAS BY VOLUME PERCENT	GAS FLOW AT STANDARD CONDITIONS[a] cfm
½	10	18.4
½	8	22.5
1	10	36.8
1	8	45.0
1½	10	55.2
1½	8	67.5

[a] 70 F and 29.92 in. Hg.

of this group of chimneys was also investigated under pick-up conditions using coal, oil, and gas in the attached heaters for several rates of flue-gas temperature rise. These results will be analyzed in another report to be issued by the *National Bureau of Standards.*

Tests were made of the several chimneys for entering flue-gas temperatures of 200 F, 600 F, and 1000 F, and for gas-flow rates of 20, 45, and 70 cfm computed for a temperature of 70 F and a barometric pressure of 29.92 in. Hg. The equivalent mass-flow rates approximate those obtained when fuel oil is burned at rates of ½, 1, and 1½ gal per hr with 8 to 10 percent carbon dioxide in the flue gases as shown in Table 1. This range of conditions was considered to cover the operating conditions likely to occur in domestic chimneys of the sizes used. The gas-flow rates used could readily be converted into equivalent coal consumption rates by applying suitable factors for the weight of air supplied per pound of coal burned.

It was desirable to compare the performance of all the chimneys under identical atmospheric conditions. Since the tests could not all be made with the same outside temperature and barometric pressure, the observations were corrected for an outdoor temperature of 32 F and a pressure of one standard atmosphere. The formula used for making these corrections was

$$S_2 - S_1 = O_2 - O_1 \quad \ldots \ldots \ldots \ldots \ldots \quad (1)$$

where

S_1 = computed static draft or theoretical draft under the conditions of the test.
S_2 = computed static draft under the standard conditions.
O_1 = observed draft under the experimental conditions.
O_2 = corrected draft that should have been observed under standard conditions.

The relation in Equation 1 develops if the friction of a given chimney remains constant for the same mass flow and the same entering temperature at the experimental conditions and at the standard conditions. All tests were made at times when there was a negligible movement of air around the chimney.

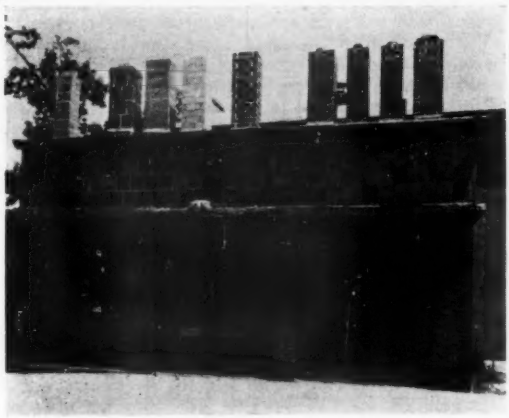


FIG. 1. TEST STRUCTURE AND CHIMNEYS

The static draft or theoretical draft of a chimney was computed from the following expression

$$D = \frac{dH\,(T_1 - T_2)}{5.2T_1} \quad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

where

D = static draft, inches water gage (in. W.G.).
d = mean density of atmospheric air at time of test, pounds per cubic foot.
H = effective height of the chimney, feet.
T_1 = mean temperature of gases in the chimney, Fahrenheit degrees absolute.
T_2 = mean temperature of outdoor air, Fahrenheit degrees absolute.

For the purpose of this computation, it was assumed that the flue gas had the same density as atmospheric air at the same temperature and pressure.

DESCRIPTION OF CHIMNEYS

Each of the 14 chimneys differed in some respect from the others as to the size, shape, or material of the flue liner; the treatment of the space between

liner and inside face of the masonry; or the material used in forming the masonry chimney wall. Each chimney was like the others in having about two thirds of its height inside a building and one third exposed above the roof.

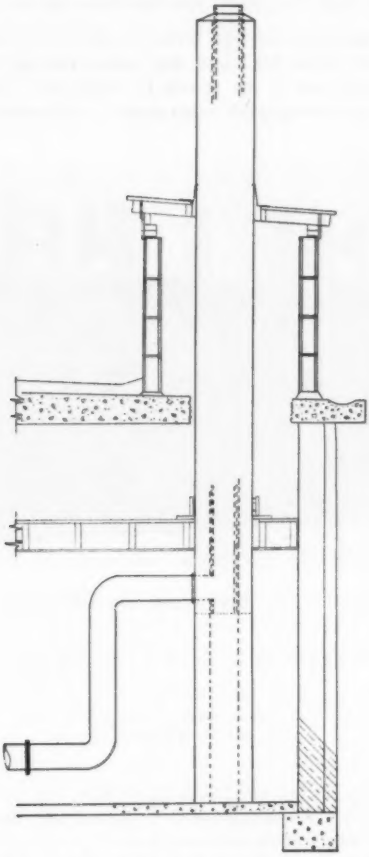


FIG. 2. ELEVATION OF TYPICAL CHIMNEY

Each had a single thick wall of masonry laid up in a mortar of 1:1:5 mix of hydrated lime, cement, and sand, and each had a round metal or tile thimble. Fig. 1 shows the structure in which the tests were made and illustrates the chimneys protruding through the roof. The height of the several chimneys from the center of the thimble to the top ranged from 15.2 to 15.6 ft, except one of the two liners in chimney 3 and this was 16.2 ft in height.

The materials used in each chimney and the treatment of the air space around the liner are described in following paragraphs. The flue liners have been designated by their nominal internal dimensions for ease in comparing the cross-sectional areas of the several specimens.

Chimney 1. Common brick having 6 core holes, 11/16 in. diam, 7- by 11-in. fire clay flue liner surrounded by air space, 8 in.-diam metal thimble.

Chimney 2. Common brick, 7- by 11-in. fire clay flue liner grouted in with mortar, 8 in.-diam metal thimble.

Chimney 3. Common brick, two 7- by 11-in. fire clay liners contiguous to each other and surrounded with an air space, and two 8 in.-diam fire clay thimbles on the same face of the chimney. This construction resulted in one flue being 16.2 ft high.

Chimney 4. Common brick, 10 in.-diam fire clay flue liner surrounded by an air space, 8 in.-diam thimble of tile.

Chimney 5. Shale multicored tile 3⅝ in. by 4 in. by 12 in. in size, 7- by 11-in. fire clay liner surrounded by an air space, 8 in.-diam thimble of tile.

Chimney 6. Fire clay multicored tile 4 in. by 5 in. by 12 in. in size, 7- by 11-in. fire clay tile liner surrounded by an air space, 8 in.-diam thimble of tile.

Chimney 7. Shale partition tile having webs and shells approximately 9/16 in. thick, 7- by 11-in. fire clay tile liner with an air space, 8 in.-diam thimble of tile.

Chimney 8. Common brick, 7 in.-diam fire clay liner with air space filled with expanded mineral matter insulation weighing about 3 lb per ft^3 for insulation, 7 in.-diam integral tile thimble.

Chimney 9. Common brick, 7 in.-diam enameled steel flue liner with air space filled with expanded mineral matter insulation, 7 in.-diam integral enameled steel thimble.

Chimney 10. Cinder concrete building block having two cells, 7- by 7-in. fire clay flue liner with an air space, 7 in.-diam thimble of tin.

Chimney 11. Same as chimney 10 except that solid cinder concrete blocks were used.

Chimney 12. Cinder concrete chimney unit blocks 17 by 22 by 8 in., designed to be used with 7- by 11-in. fire clay liner. Except for a spacing boss at the midlength of each side, the liner was spaced ¾ in. from the interior of the unit.

Chimney 13. Cinder concrete chimney unit blocks 16 by 16 by 8 in., 7- by 7-in. fire clay liner grouted in solid with mortar, 7 in.-diam thimble of tile.

Chimney 14. Same as chimney 13 except that the liner was grouted only at the joints.

An elevated view of a typical chimney is shown in Fig. 2. It will be noted in this figure that all but the top 5½ ft of the chimney was surrounded by air at room temperature. Fig. 3 is a photograph of the lower parts of chimneys 6, 7 and 12 showing the different materials of construction above the thimbles. The bases of the chimneys shown were not of the same materials as the portion above the thimbles.

TEST EQUIPMENT AND PROCEDURE

The method used for the tests consisted of blowing air through a calibrated orifice, then over a gas burner into the chimney. The orifice was used for measuring the volume of air and the volume of gas used was measured with a dry gas meter. A small centrifugal blower was used to overcome the pressure drop of the orifice. The system was sealed against extraneous air leakage and the pipe was insulated with 4-in. rock-wool batts from the burner to the chimney to obtain nearly uniform flue-gas temperature across the pipe. A view of one

of the sets of equipment is shown in Fig. 4 in a typical position attached to chimney 6. As there were 7- and 8 in.-diam thimbles in the test chimneys, two duplicate sets of equipment were constructed, to permit simultaneous tests of two specimens.

The heater consisted of a single gas burner built into a section of smokepipe with the burner exactly in the center of the pipe. The calibrated orifice plates

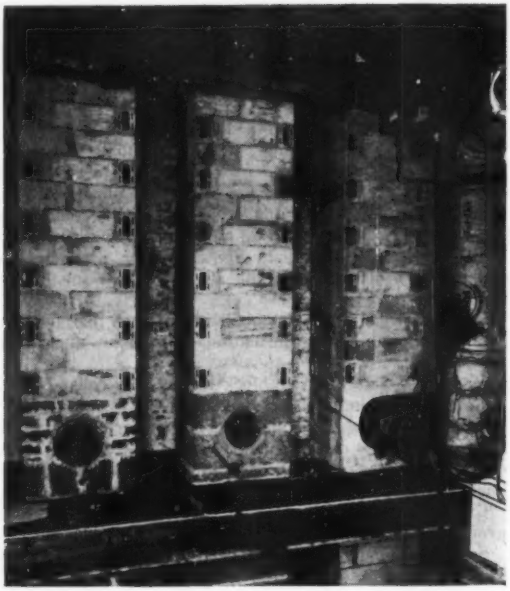


FIG. 3. VIEW OF CHIMNEYS 6, 7 AND 12 SHOWING BASES, THIMBLES AND CONSTRUCTION ABOVE BASE

were bolted between two pieces of flanged pipe so that the orifices could be exchanged without disturbing the remainder of the equipment. The pressure drop across the orifice was measured with an inclined draft gage graduated in thousandths inch water gage (W.G.).

The draft tube entered the smokepipe 1 ft from the face of the chimney. The axis of the tube was normal to the flow of gases. The draft was observed on a calibrated inclined gage graduated in thousandths inch water gage (W.G.). All inclined gages were calibrated with a Wahlen differential manometer after rigid mounting on the brick wall of the building.

Temperatures in the chimney and smokepipe were measured with unshielded thermocouples of chromel-alumel wire, and an automatic, indicating, electronic potentiometer which could easily be read to 1 deg. Thermocouples were inserted

through sleeves in the chimney and smokepipe to the center of the gas stream in the following stations:

Station 1. In the smokepipe in a plane 1 ft from the chimney face, but at right angles to the draft tube.
Station 2. In the chimney about 1 ft above the center of the thimble.
Station 3. In the chimney midway between ceiling and roof.
Station 4. In the chimney at the roof level.
Station 5. In the chimney 6 in. below the top of the flue.

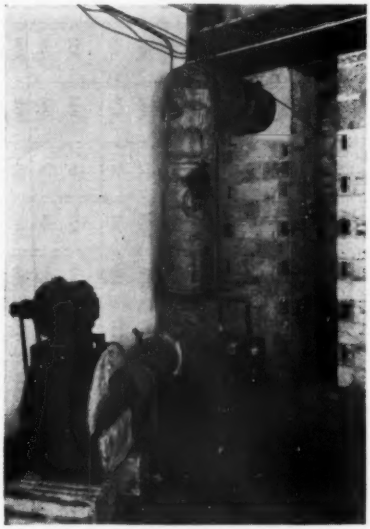


FIG. 4. TEST EQUIPMENT ATTACHED TO CHIMNEY

Thermocouples were also used to measure the outside air temperature and the ambient temperature inside the test structure.

The chimneys were tested in pairs with the same mass flow of gases and the same flue-gas temperature being used in both. The inlet air and gas consumption rate was regulated until the desired flue-gas temperature and volume were obtained. These conditions were then maintained for a period of 6 hr, after which all observations were recorded at intervals of 15 min for a period of 1 hr.

TEST RESULTS

Chimney Draft

The principal data obtained on the 14 chimneys are summarized in Table 2. A plan view of each chimney is shown in Table 2 opposite the data obtained therefrom illustrating the materials and method of construction. This table

TABLE 2. PERFORMANCE OF MASONRY CHIMNEYS UNDER STEADY STATE CONDITIONS

Chimney	Nominal flue-gas flow rate cfm	Observed flue-gas flow rate cfm	Observed flue-gas Temperature					Average flue-gas temp °F	Outside temp °F	Draft at test conditions		Draft at Standard conditions[a]		Indicated friction loss in. W.G.	Chimney effectiveness Percent
			Station 1 at inlet °F	Station 2 °F	Station 3 °F	Station 4 °F	Station 5 °F			Observed at chimney inlet in. W.G.	Computed static in. W.G.	Observed in. W.G.	Computed static in. W.G.		
Common Brick No. 1 15.3 ft 17" × 21" 7"x11" liner	20	20.3	201	159	136	131	109	131	68	0.023	0.023	0.040	0.040	0.000	65.6
	20	20.7	601	429	330	310	225	312	77	.070	.065	.090	.085	−.005	70.7
	20	21.4	999	680	477	477	270	459	57	.102	.099	.113	.110	−.003	71.9
	45	43.0	199	172	167	158	145	159	73	.028	.030	.047	.049	+.002	78.3
	45	43.6	601	462	444	404	342	410	71	.088	.085	.106	.103	−.003	83.3
	45	44.4	1000	760	731	650	540	662	77	.114	.113	.134	.133	−.001	85.2
	70	70.0	199	162	157	151	141	152	62	.029	.033	.043	.047	+.004	71.7
	70	70.9	601	507	495	475	424	473	72	.091	.094	.109	.112	+.003	85.7
	70	72.2	1000	843	809	757	656	761	57	.129	.128	.143	.142	−.001	90.9
Common Brick No. 2 15.5 ft 17" × 21" 7"x11" liner	20	21.3	200	155	147	119	101	127	68	.023	.022	.040	.039	−.001	65.6
	20	21.6	601	419	380	276	190	304	77	.066	.064	.087	.085	−.002	67.4
	20	21.6	995	667	579	430	266	465	57	.100	.099	.113	.112	−.001	71.9
	45	43.5	199	176	173	153	138	158	73	.028	.030	.047	.049	+.002	77.0
	45	44.0	602	471	459	375	310	396	71	.086	.084	.104	.102	−.002	80.5
	45	44.7	999	781	729	598	489	633	77	.113	.112	.133	.132	−.001	83.5
	70	70.2	199	176	175	160	147	163	62	.032	.037	.046	.051	+.005	75.4
	70	71.2	601	525	510	459	415	472	72	.091	.095	.109	.113	+.004	84.4
	70	72.4	1000	857	814	729	640	751	57	.131	.129	.144	.142	−.002	90.4

TABLE 2 *(continued)*. PERFORMANCE OF MASONRY CHIMNEYS UNDER STEADY STATE CONDITIONS

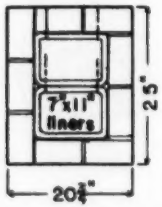
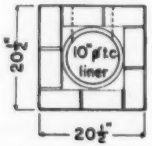
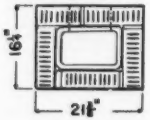
Common Brick  No. 3 15.2 ft	20	20.1	200	158	138	120	109	128	70	0.021	0.021	0.039	0.039	0.000	65.0
	20	20.4	600	417	329	241	198	278	66	.063	.063	.079	.079	.000	62.6
	20	20.8	1000	669	503	356	279	421	73	.083	.086	.102	.105	+.003	65.3
	45	45.2	200	170	168	150	144	156	87	.021	.024	.045	.048	+.003	75.0
	45	45.8	601	450	406	334	276	364	72	.073	.077	.091	.095	+.004	71.5
	45	46.6	1000	762	668	528	420	572	65	.102	.108	.116	.122	+.006	74.2
	70	70.4	199	173	172	154	146	160	54	.031	.038	.042	.049	+.007	70.0
	70	71.4	601	489	473	394	347	418	64	.082	.090	.096	.104	+.008	75.5
	70	72.5	1000	798	753	614	542	662	64	.111	.118	.126	.133	+.007	80.6
Common Brick  No. 4 15.3 ft	20	20.0	199	162	162	140	124	145	70	.027	.027	.039	.039	.000	65.0
	20	20.4	600	426	423	340	262	358	66	.079	.079	.094	.094	.000	73.9
	20	20.3	1001	689	651	508	376	544	73	.107	.102	.126	.121	−.005	80.1
	45	45.2	200	180	180	171	163	173	87	.029	.029	.053	.053	.000	86.8
	45	45.8	600	477	474	426	367	434	72	.089	.088	.107	.106	−.001	84.1
	45	46.6	1000	811	778	687	577	700	65	.122	.120	.138	.136	−.002	87.8
	70	70.0	199	184	183	175	162	176	54	.038	.043	.049	.054	+.005	81.7
	70	71.0	601	520	511	476	428	481	64	.100	.100	.113	.113	.000	88.8
	70	72.1	1000	856	823	751	654	764	64	.130	.128	.145	.143	−.002	92.2
3⅜"x4"x12" Shale Multicored Tile  No. 5 15.5 ft	20	20.3	199	165	158	136	122	143	72	.026	.026	.044	.044	.000	72.2
	20	20.5	601	450	404	302	216	332	72	.072	.073	.090	.091	+.001	69.6
	20	20.9	1000	702	633	453	305	506	86	.092	.093	.117	.118	+.001	73.5
	45	45.2	199	178	178	165	154	168	81	.029	.030	.051	.052	+.001	83.6
	45	45.9	601	498	487	424	363	439	85	.085	.085	.109	.109	.000	84.4
	45	46.7	1000	824	776	659	549	695	82	.112	.114	.136	.138	+.002	85.4
	70	70.0	200	180	180	171	165	174	86	.028	.030	.052	.054	+.002	85.2
	70	71.0	601	509	505	450	401	460	81	.089	.089	.112	.112	.000	86.7
	70	72.3	1000	855	819	726	632	747	83	.117	.119	.140	.172	+.002	87.9

TABLE 2 *(continued)*. PERFORMANCE OF MASONRY CHIMNEYS UNDER STEADY STATE CONDITIONS

Chimney	Nominal flue-gas flow rate cfm	Observed flue-gas flow rate cfm	Observed flue-gas Temperature					Average flue-gas temp °F	Outside temp °F	Draft at test conditions		Draft at Standard conditions*		Indicated friction loss in. W.G.	Chimney effectiveness Percent
			Station 1 at inlet °F	Station 2 °F	Station 3 °F	Station 4 °F	Station 5 °F			Observed at chimney inlet in. W.G.	Computed static in. W.G.	Observed in. W.G.	Computed static in. W.G.		
No. 6 4"x5"x12" Fire Clay Multicored Tile 15.6 ft 16" 7" x 11" liner 21⅛"	20	20.1	199	147	161	138	124	142	72	0.026	0.026	0.044	0.044	0.000	72.1
	20	20.5	601	392	391	305	232	324	72	.072	.071	.092	.091	−.001	70.6
	20	20.9	1000	658	622	460	337	507	86	.094	.093	.119	.118	−.001	74.2
	45	45.4	199	170	175	162	155	165	81	.028	.030	.050	.052	+.002	82.0
	45	46.0	601	477	485	415	382	436	85	.085	.085	.109	.109	.000	83.7
	45	46.6	1000	795	773	640	577	686	82	.112	.114	.136	.138	+.002	84.9
	70	70.4	200	176	180	168	166	170	86	.027	.029	.051	.053	+.002	82.2
	70	71.4	601	418	500	444	416	457	81	.089	.089	.113	.113	.000	86.8
	70	72.5	1000	823	818	708	651	741	83	.116	.118	.140	.142	+.002	87.5
No. 7 4"x12"x12" Shale Tile 15.5 ft 16" 7"x11" liner 21⅛"	20	20.2	199	154	154	119	111	131	69	.024	.023	.042	.041	−.001	68.8
	20	20.5	601	407	396	273	226	315	86	.071	.064	.095	.088	−.007	73.5
	20	20.9	1000	675	622	412	320	480	90	.095	.090	.112	.107	−.005	70.3
	45	45.2	199	166	176	162	159	166	87	.027	.027	.051	.051	.000	83.6
	45	45.9	601	449	473	383	376	416	89	.082	.079	.108	.105	−.003	83.6
	45	46.7	1000	757	755	577	561	648	80	.115	.115	.133	.133	.000	83.5
	70	70.3	199	163	169	160	158	163	79	.029	.030	.050	.051	+.001	82.0
	70	71.2	601	457	482	424	403	440	82	.085	.086	.108	.109	+.001	83.6
	70	72.4	1000	787	793	680	640	718	72	.119	.121	.138	.140	+.002	86.7

TABLE 2 *(continued)*. PERFORMANCE OF MASONRY CHIMNEYS UNDER STEADY STATE CONDITIONS

Common Brick No. 8 15.5 ft	20	20.3	199	174	171	151	136	157	55	0.035	0.038	0.046	0.049	+.003	75.4
	20	20.4	601	491	467	404	349	424	72	.085	.088	.104	.107	+.003	80.5
	20	21.0	1000	802	723	619	509	659	54	.119	.123	.130	.134	+.004	81.8
	45	45.1	199	194	187	177	169	181	67	.034	.039	.051	.056	+.005	83.6
	45	45.7	601	541	505	462	425	478	63	.093	.100	.107	.114	+.007	82.8
	45	46.6	1000	893	836	763	687	795	56	.121	.135	.133	.147	+.014	83.5
	70	70.0	199	189	185	180	173	181	73	.027	.037	.046	.056	+.010	75.4
	70	71.3	602	570	548	517	482	527	50	.093	.113	.102	.122	+.020	79.0
	70	72.4	1000	923	884	828	759	844	53	113	.140	.124	.151	+.027	77.9
Common Brick No. 9 15.6 ft	20	20.2	199	183	180	163	142	167	55	.037	.041	.048	.052	+.004	78 7
	20	20.4	601	518	507	446	371	459	72	.093	.094	.111	.112	+ 001	85.2
	20	20.8	1000	860	818	708	548	732	54	.128	.131	.139	.142	+.003	86.8
	45	45.2	199	188	188	181	171	182	67	.034	.040	.050	.056	+.006	81 8
	45	45.8	600	549	545	509	445	512	63	.094	.105	.110	.121	+.011	84.5
	45	46.6	1000	913	896	823	702	834	56	.123	.139	.135	.151	+.016	84.2
	70	70.3	199	190	190	186	176	186	73	.027	.039	.046	.058	+.012	75.4
	70	71.4	602	576	570	539	486	543	50	.094	.114	.103	.123	+.020	79.1
	70	72.3	1000	947	932	875	768	881	53	.115	.143	.126	.154	+.028	78.6
4"x8"x12" Cinder Blocks No. 10 15.3 ft	20	19.9	199	164	162	133	108	140	34	.033	.042	.034	.043	+.009	55.7
	20	20.4	601	467	444	353	260	374	32	.093	.098	.093	.098	+.005	73.1
	20	21.0	1000	786	711	558	404	600	49	.114	.120	.121	.127	+.006	76.9
	45	45.2	199	183	181	168	151	170	44	.039	.047	.044	.052	+.008	72.1
	45	45.9	603	521	506	448	371	458	46	.094	.104	.101	.111	+.010	78.8
	45	46.7	999	879	830	744	612	760	50	.120	.134	128	.142	+.014	81.4
	70	70.4	199	190	187	177	165	179	36	.039	053	.041	.055	+.014	67.2
	70	71.3	601	542	525	490	419	492	59	.088	.101	.103	.116	+.013	81.0
	70	72.4	1000	906	864	809	702	815	71	.112	.124	.132	.144	+.012	83.9

TABLE 2 *(continued)*. PERFORMANCE OF MASONRY CHIMNEYS UNDER STEADY STATE CONDITIONS

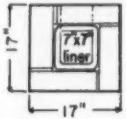
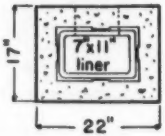
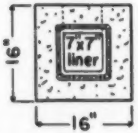
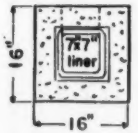
Chimney	Nominal flue-gas flow rate cfm	Observed flue-gas flow rate cfm	Observed flue-gas Temperature					Average flue-gas temp °F	Outside temp °F	Draft at test conditions		Draft at Standard conditions[a]		Indicated friction loss in. W.G.	Chimney effectiveness Percent
			Station 1 at inlet °F	Station 2 °F	Station 3 °F	Station 4 °F	Station 5 °F			Observed at chimney inlet in. W.G.	Computed static in. W.G.	Observed in. W.G.	Computed static in. W.G.		
4"x8"x12" Cinder Blocks No. 11 15.3 ft 	20	20.2	199	166	158	136	108	140	34	0.033	0.042	0.034	0.043	+.009	55.7
	20	20.7	600	470	429	357	257	368	32	.094	.096	.094	.096	+.002	73.9
	20	20.9	1000	792	689	565	404	594	49	.113	.119	.121	.127	+.006	77.0
	45	44.8	199	190	184	176	158	176	44	.040	.048	.045	.053	+.008	73.8
	45	45.8	603	532	504	467	389	470	46	.096	.105	.102	.111	+.009	79.5
	45	46.7	999	877	824	757	634	767	50	.122	.134	.130	.142	+.012	82.7
	70	70.3	199	184	182	178	165	178	36	.039	.052	.041	.054	+.013	67.2
	70	71.1	601	543	523	496	437	498	59	.091	.101	.105	.115	+.010	82.6
	70	72.3	1000	904	852	815	722	820	71	.113	.124	.135	.146	+.011	85.9
17"x22"x8" Cinder Concrete Chimney Blocks No. 12 15.5 ft 	20	20.2	199	156	151	133	113	137	69	.023	.025	.040	.042	+.002	65.6
	20	20.5	601	410	362	306	253	325	86	.068	.066	.092	.090	−.002	71.2
	20	21.0	999	681	579	449	359	499	90	.092	.088	.121	.117	−.004	76.0
	45	45.3	199	176	177	165	161	169	87	.027	.028	.052	.053	+.001	85.2
	45	45.9	601	478	470	405	374	427	89	.081	.082	.106	.107	+.001	82.0
	45	46.7	1000	791	745	622	554	661	80	.113	.113	.135	.135	.000	84.8
	70	69.9	199	181	181	170	168	174	79	.033	.032	.055	.054	−.001	90.1
	70	70.8	601	493	479	418	404	442	82	.085	.087	.108	.110	+.002	83.6
	70	72.0	1000	806	779	666	638	711	72	.119	.121	.138	.140	+.002	86.6

TABLE 2 *(concluded)*. PERFORMANCE OF MASONRY CHIMNEYS UNDER STEADY STATE CONDITIONS

16"x16"x8" Cinder Concrete Chimney Blocks No. 13 15.5 ft 	20	20.1	200	168	164	141	130	149	72	0.027	0.028	0.045	0.046	+.001	73.8
	20	20.4	601	455	430	340	285	368	59	.083	.085	.096	.098	+.002	74.3
	20	21.0	1000	754	673	516	417	574	61	.108	.112	.123	.127	+.004	77.2
	45	45.0	199	180	178	167	160	170	73	.028	.034	.047	.053	+.006	77.0
	45	45.5	601	520	495	437	402	457	67	.093	.096	.109	.112	+.003	84.4
	45	46.6	1000	851	796	690	624	727	81	.114	.118	.137	.141	+.004	86.0
	70	70.2	199	176	175	167	164	170	77	.028	.032	.047	.051	+.004	77.0
	70	71.2	601	534	518	474	452	490	79	.090	.094	.112	.116	+.004	86.7
	70	72.3	1000	880	842	761	718	790	82	.117	.123	.140	.146	+.006	87.9
16"x16"x8" Cinder Concrete Chimney Blocks No. 14 15.5 ft 	20	20.1	200	166	162	142	134	149	72	.027	.028	.045	.046	+.001	73.8
	20	20.4	601	456	417	328	271	359	59	.080	.083	.093	.096	+.003	72.0
	20	20.8	1000	752	649	495	401	552	61	.104	.109	.119	.124	+.005	74.7
	45	45.2	199	181	180	174	163	177	73	.029	.036	.048	.055	+.007	78.7
	45	45.9	601	505	481	434	386	446	67	.088	.094	.104	.110	+.006	80.5
	45	46.8	1000	833	775	688	602	714	81	.109	.116	.132	.139	+.007	82.9
	70	70.2	199	187	185	181	175	182	77	.029	.034	.051	.056	+.005	83.6
	70	71.2	601	531	510	485	446	490	79	.085	.094	.107	.116	+.009	82.8
	70	72.5	1000	874	828	772	698	786	82	.111	.122	.135	.146	+.011	84.8

[a] Standard conditions; barometric pressure 29.92 in. Hg., outdoor temperature 32 F.

shows the temperatures observed at five stations in each chimney, the draft observed during the test, the corrected draft for an outdoor temperature of 32 F and a barometric pressure of 29.92 in. Hg, the indicated friction loss, and the chimney effectiveness for three entering flue-gas temperatures and three rates of flue-gas flow. An examination of Table 2 shows that the draft produced by the several chimneys under steady state conditions did not differ greatly

TABLE 3. CLASSIFICATION OF CHIMNEYS AS TO MASONRY MATERIALS

CHIMNEY		
COMMON BRICK	TILE	CINDER-CONCRETE BLOCK
Specimen	Specimen	Specimen
1	5	10
2	6	11
3	7	12
4	—	13
8	—	14
9	—	—

TABLE 4. CLASSIFICATION OF CHIMNEYS AS TO SIZE AND SHAPE OF LINER

CHIMNEY			
RECTANGULAR 7 BY 11 IN. AREA 77 SQ IN.	ROUND DIAM 10 IN. AREA 78.5 SQ IN.	ROUND DIAM 7 IN. AREA 38.5 SQ IN.	SQUARE 7 BY 7 IN. AREA 49 SQ IN.
Specimen	Specimen	Specimen	Specimen
1	4	8	10
2	—	9	11
3	—	—	13
5	—	—	14
6	—	—	—
7	—	—	—
12	—	—	—

with the type of masonry material used, the type of liner used, or the treatment of the air space. However, specific pairs or groups of chimneys have been selected for comparison to show the magnitude of the differences observed.

The 14 chimneys are classified in Table 3 as to the kind of masonry units used for the chimney wall, and in Table 4 as to the size and shape of the liner. If the performance of chimneys 1, 2, 5, 6, 7, and 12, using different masonry materials, but all having a 7- by 11-in. liner, is compared, it will be noted that the observed draft, when corrected to standard conditions, varied less than 0.01 in. W.G. at all flue-gas temperatures and volumes with one exception for the six chimneys. The variation among the drafts produced by these six chimneys was 10 percent or less, except for the case of chimneys 1 and 12 when flue gases entered at a temperature of 200 F and a rate of 70 cfm. In this case, the

observed draft at standard conditions was 0.043 in. W.G. for chimney 1 and 0.055 in. W.G. for chimney 12, a difference of 25 percent.

The comparison of a metal lining of enameled steel and a refractory lining in otherwise similar chimneys is afforded by specimens 8 and 9. Table 2 shows that the performances of the two chimneys were nearly identical for gas flow rates of 45 and 70 cfm, but for a flow of 20 cfm the metal-lined chimney produced slightly more draft at all three temperatures of the inlet gases. These two chimneys were tested simultaneously so that both were subjected to the same

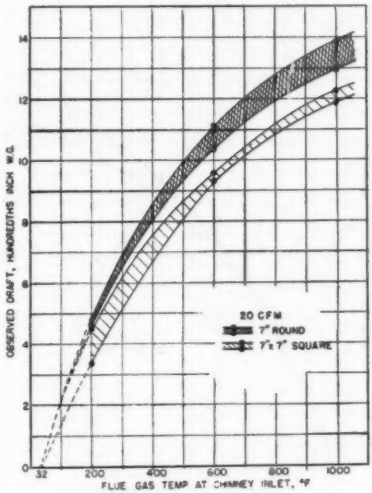


FIG. 5. RANGE OF OBSERVED DRAFT IN ROUND AND SQUARE CHIMNEYS (20 CFM FLOW RATE)

Outside temperature, 32 F. Barometric pressure 29.92 in. Hg.

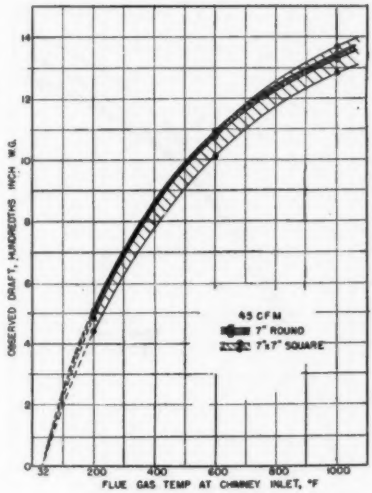


FIG. 6. RANGE OF OBSERVED DRAFT IN ROUND AND SQUARE CHIMNEYS (45 CFM FLOW RATE)

Outside temperature, 32 F. Barometric pressure 29.92 in. Hg.

atmospheric conditions during the tests. Table 2 also shows that the indicated friction losses in the two specimens were almost equal and that the chimney effectiveness varied slightly.

The range of draft observed in the two chimneys with 7-in. round liners and the four chimneys with 7- by 7-in. square liners is shown on Figs. 5, 6, and 7. In capacity, liners of these two shapes are sometimes considered to be comparable. Fig. 5 shows that the chimneys with the round liners produced appreciably more draft than those with the square liners for a gas flow of 20 cfm. Fig. 6 shows that the performances of the two types of liners were approximately equivalent for a gas flow of 45 cfm. Fig. 7 shows that the square liners were superior to the round liners for a gas flow of 70 cfm when the flue-gas temperature was 500 F or higher. This suggests that the two linings are not

equivalent except possibly for gas flows near 45 cfm, *i.e.*, when 1 gal of fuel oil is burned per hour with 8 percent carbon dioxide. It further suggests that the insulation used around the two circular liners may have benefited these two chimneys for low rates of gas flow, but that the greater area of the square liner more than offset the effect of the insulation at rates of flow near 70 cfm. The available draft successively increased in the square liner as the gas flow was increased from 20 to 70 cfm, whereas it decreased in the round liners as the gas flow was increased from 45 to 70 cfm.

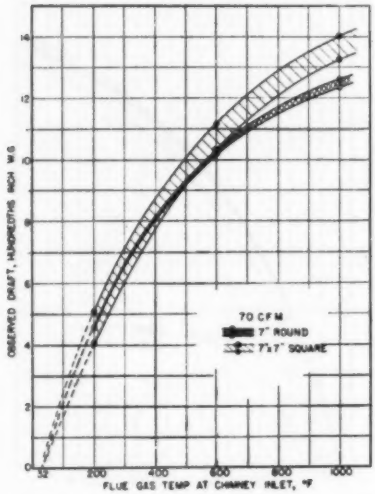


FIG. 7. RANGE OF OBSERVED DRAFT IN ROUND AND SQUARE CHIMNEYS (70 CFM FLOW RATE)

Outside temperature, 32 F. Barometric pressure 29.92 in. Hg.

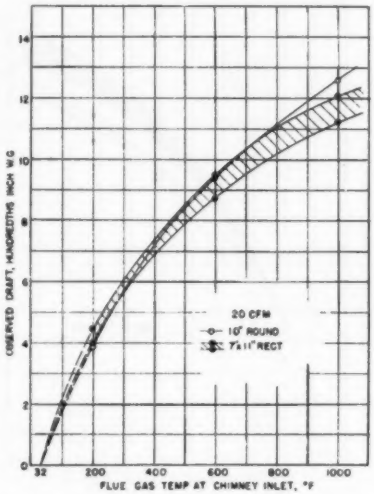


FIG. 8. RANGE OF OBSERVED DRAFT IN ROUND AND RECTANGULAR CHIMNEYS (20 CFM FLOW RATE)

Outside temperature, 32 F. Barometric pressure 29.92 in. Hg.

A similar comparison is made in Figs. 8, 9, and 10, between the one chimney having a 10-in. circular liner and the six chimneys having 7- by 11-in. rectangular liners. Chimney 3 with two 7- by 11-in. liners is not included in this comparison. The 10-in. circular liner and the 7- by 11-in. liners have practically the same internal cross-sectional area. It will be noted in Figs. 8, 9, and 10 that the draft produced in the round liner was within the range of drafts observed in the six chimneys with rectangular liners for all conditions, except the highest entering flue-gas temperature plotted in Fig. 8. At entering flue-gas temperatures of 1000 F, the draft observed in the round liner was equal to, or exceeded, the highest value observed in the 7- by 11-in. liners. The available draft increased steadily with increased gas flow in the range from 20 to 70 cfm.

For the same conditions of gas flow and flue-gas temperature, the chimneys with 7- by 7-in. liners produced almost the same amount of available draft as those with 7- by 11-in. liners in the range used for these tests. The data in Table 2 show that the average flue-gas temperature was somewhat higher in the 7- by 7-in. liners than for the larger liners with the same inlet conditions, but the loss of draft due to friction was also measurably greater in the 7-in. flue lining so that the available draft was nearly equal for the two sizes of chimney

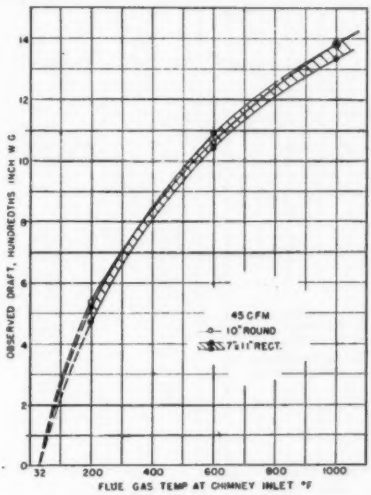


FIG. 9. RANGE OF OBSERVED DRAFT IN ROUND AND RECTANGULAR CHIMNEYS (45 CFM FLOW RATE)

Outside temperature, 32 F. Barometric pressure 29.92 in. Hg.

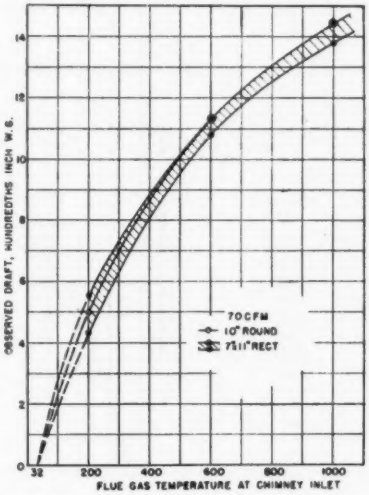


FIG. 10. RANGE OF OBSERVED DRAFT IN ROUND AND RECTANGULAR CHIMNEYS (70 CFM FLOW RATE)

Outside temperature, 32 F. Barometric pressure 29.92 in. Hg.

liner. It is probable that at some flue-gas flow rate above 70 cfm the larger flue lining would provide appreciably more draft than the 7- by 7-in. flue liner.

The results obtained on chimney 3 with two 7- by 11-in. clay tile liners for a flue gas rate of 45 cfm are plotted on Fig. 11. Tests were made with gases entering both thimbles in the same quantity and at the same temperature and also with hot gases entering only the upper thimble while the lower thimble and corresponding liner were left open. Fig. 11 shows that from 0.005 to 0.017 in. W.G. less draft was produced by the liner served by the upper thimble when the other liner was open than when both liners were carrying hot gases. This decrease in draft amounted to 10 to 15 percent of the total available draft. The back liner, served by the lower thimble, produced more draft than the other liner presumably because it was greater in length by 1 ft. A comparison of the drafts shown in Table 2 for chimneys 1, 2, 5, 6, 7, and 12 having single 7-

by 11-in. liners with the drafts shown on Fig. 11 for the front liner in chimney 3, when both liners in that chimney were supplied with hot flue gases, shows that most of the other chimneys produced a little more draft than did the comparable liner in chimney 3. These differences ranged from 0.000 to 0.006 in. W.G.

Chimney 1 was tested with the base of the chimney below the thimble sealed off and with the base open for comparison. A flue-gas rate of 70 cfm was used for these tests. The maximum difference in available draft observed for any

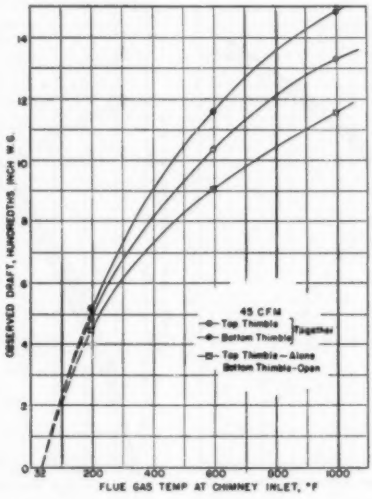


FIG. 11. OBSERVED DRAFT IN CHIMNEY 3 (TWO FLUES, 7 IN. X 11 IN.)
Outside temperature, 32 F. Barometric pressure 29.92 in. Hg.

flue-gas temperature was 0.004 in. W.G. or about 3 percent. This variation was considered small and probably of the same order as the experimental error for the test method used. Openings in the masonry wall and lining of the chimney near the base with an area of one percent of the interior chimney area made no observable difference in the available draft produced by the chimney with the base open. Undoubtedly, large openings or cracks in the masonry and lining would lower the draft, and. in such cases sealing the base would improve the performance of an existing chimney.

Comparisons of the performance of chimneys with the linings grouted in solid with those having only the liner joints grouted are provided by chimneys 1 and 2 and chimneys 13 and 14. In each of these pairs, the chimneys were of identical construction and material, except for the treatment of the air space. A comparison of the data for chimneys 1 and 2 in Table 2 shows almost identical performance. The maximum difference in draft for the two chimneys was

0.004 in. W.G. with a flue-gas volume of 20 cfm and an entering temperature of 600 F.

The drafts observed in chimney 13 exceeded those in chimney 14 by amounts ranging from 0.002 in. to 0.005 in. W.G., for flue-gas temperatures from 600 F to 1000 F. This is the opposite effect from that expected if the open air space had any appreciable insulating effect.

The data in Table 5, excerpted from Table 2, show that the average flue-gas temperature in chimney 13 with a grouted liner was higher than that in chimney 14 with an air space for entering temperatures of 600 F and 1000 F,

TABLE 5. TEMPERATURES, DRAFT, AND FRICTION IN CHIMNEYS 13 AND 14

Flue-Gas Temperature at Inlet F	Average Flue-Gas Temperature F		Differences in Computed Static Draft in. W.G.	Computed Draft Loss Due to Friction in. W.G.	
	Chimney 13	Chimney 14	Chimneys 13 and 14	Chimney 13	Chimney 14
200	149	149	0.000	0.001	0.001
601	368	359	+ .002	.002	.003
1000	574	552	+ .003	.004	.005
199	170	177	− .002	.006	.007
601	457	446	+ .002	.003	.006
1000	727	714	+ .002	.004	.007
199	170	182	− .002	.004	.005
601	490	490	.000	.004	.009
1000	790	786	+ .001	.006	.011

but that the computed static draft corresponding to these temperatures did not differ by more than 0.003 in. W.G. for any test.

Either of the following conditions might be the reason for the observed equivalence of similar chimneys with and without an air space. An open air space would permit leakage of cold air through the outside masonry if there were any cracks in it, or it would permit leakage of hot gases from the liner if there were any cracks in the liner or between joints. Either kind of leakage would obviate the insulating effect of the air space. The cinder-concrete blocks used in chimneys 13 and 14 appeared somewhat porous, hence, leakage through the walls themselves would also be expected in the chimney with an air space such as chimney 14.

Chimney Temperatures: Table 2 shows that the average temperature of the gases in the chimneys increased as the mass flow increased. This result was to be expected since the heat loss of a given chimney would be approximately constant for a given entering flue-gas temperature and consequently a smaller temperature drop would occur with a greater mass flow.

Table 2 also shows that in several of the chimneys, the observed flue-gas temperature at station 3, midway between the ceiling and roof, was as high as that at station 2, approximately 1 ft above the chimney inlet. This result is attributed to distortion of the gas stream at the chimney entrance caused by the change of direction and possibly by down-drafts. The temperatures were measured at the center of the flue-gas stream. It is probable that the maximum

temperatures did not always occur at the center of the stream near the chimney entrance.

The data in Table 2 show that the average flue-gas temperature decreased in the following order for the liners: 7-in. round liner, 7- by 7-in. square liner, 7- by 11-in. rectangular liner. There was no consistent difference between the average flue-gas temperatures in the 10-in. round liner and the several 7- by 11-in. liners. That is, the average temperatures observed in the round liner were within the range of average temperatures observed in the six 7- by 11-in. liners. Theoretically, it is to be expected that the heat loss of a chimney would be approximately proportional to the interior perimeter of the liner for chimneys

TABLE 6. COMPARISON OF OBSERVED DRAFT AT STANDARD CONDITIONS FOR GROUP OF CHIMNEYS DESCRIBED IN THIS REPORT WITH TWO BRICK CHIMNEYS PREVIOUSLY TESTED

Liner, 7 by 7 in.			Liner, 7 by 11 in.		
Inlet Flue-Gas Temp F	Brick Chimney Previously Tested in. W.G.	Cinder-Concrete Block Chimneys (10, 11, 13, 14) in. W.G.	Inlet Flue-Gas Temp F	Brick Chimney Previously Tested in. W.G.	Brick, Shale-Tile, and Cinder-Concrete Block Chimneys (1, 2, 5, 6, 7, 12) in. W.G.
		20 cfm			
250	0.053	0.044–.053	275	0.055	0.053–.057
400	.072	.067–.075	——	——	——
600	.086	.093–.096	600	.088	.087–.095
1000	.115	.119–.123	1000	.108	.112–.121
		70 cfm			
200	.053	.041–.051	200	.052	.043–.055
400	.079	.078–.087	——	——	——
600	.102	.103–.112	600	.102	.108–.113
1000	.127	.133–.140	1000	.131	.138–.144

of similar wall thickness and similar materials. Except for the 10-in. round liner, the heat loss of the specimen chimneys conformed approximately to this pattern.

The data reported in Table 2 for chimney 3, with two adjacent liners, were obtained with flue gases entering the front and shorter liner while the rear liner was open and not connected to a heating device. The average flue-gas temperature in chimney 3 under these conditions was 7 deg lower at chimney inlet temperatures of 200 F and 78 deg lower at chimney inlet temperatures of 1000 F than the average in the other specimens with 7- by 11-in. liners.

Chimney Effectiveness: The chimney effectiveness was computed for each chimney for each different test condition, as recorded in Table 2. The chimney effectiveness is defined as the ratio of the observed draft at standard conditions to the theoretical draft that the same chimney would produce at standard atmospheric conditions with the same inlet temperature if the flue gases did not

cool as they traversed the chimney. The tabulated results show that, in most instances, the effectiveness increased as the entering flue-gas temperature increased, and as the mass flow increased in the larger chimneys. In the chimneys with 7- by 11-in. liners, and 10 in.-diam liner, the effectiveness increased successively as the flow rate was increased from 20 cfm to 45 cfm and from 45 cfm to 70 cfm. In the chimneys with 7- by 7-in. liners, the effectiveness increased for rates of flow up to 45 cfm, but did not always increase further when a flow rate of 70 cfm was used. In both 7-in. round liners, the effectiveness was lower for a flow rate of 70 cfm than for either of the two lower flow rates.

The effectiveness ranged from a low value of 65 percent to a high value of 92 percent for the chimneys with 7- by 11-in. liners and for the one with a 10-in. round liner; the minimum and maximum values of effectiveness were 56 percent and 88 percent for the specimens with 7- by 7-in. liners; and the minimum and maximum values of effectiveness were 75 percent and 87 percent for the two chimneys with 7-in. round liners.

The friction in a chimney reduces its effectiveness because it decreases the available draft at the chimney inlet. Table 2 shows that the friction was primarily responsible for the lower effectiveness of chimneys 8 and 9 when a flow rate of 70 cfm was used, whereas the cooling of the flue gases was the important factor in the larger chimneys. In chimneys 8 and 9, the indicated friction loss reached a maximum of 18 percent of the computed static draft at standard conditions for the highest flow rate.

The friction was very small in the chimneys with 7- by 11-in. liners and with the 10 in.-diam liner at all of the rates of flue-gas flow used. In general, the friction loss in those chimneys was so small that the computations sometimes gave positive values and at other times negative values, indicating that the experimental errors of the tests were greater than the values sought. The indicated friction loss was always a positive value for the chimneys with the 7-in.-diam liners and with the 7- by 7-in. square liners.

DISCUSSION AND CONCLUSIONS

The results obtained on those chimneys with 7- by 7-in. square liners and 7- by 11-in. rectangular liners were compared with results previously obtained at the *National Bureau of Standards* in tests on brick chimneys[1] with liners of these same two sizes, published in the 1948 issue of the HEATING, VENTILATING, AIR CONDITIONING GUIDE. The comparison is shown in Table 6. It will be noted that the observed drafts at standard conditions in the two brick chimneys tested several years ago were in the range of drafts observed in the chimneys with the same size liner during the later tests for inlet flue-gas temperatures from 200 F to about 600 F. For inlet flue-gas temperatures above 600 F the brick, shale-tile, and cinder-concrete block chimneys produced approximately 6 percent more draft at the chimney inlet.

The difference in the exposure of the two groups of chimneys may be the reason for the observed variation in performance. The later chimneys were exposed to the outside air for only about one-third of their height, as shown in

[1] Observed Performance of Some Experimental Chimneys, by R. S. Dill, P. R. Achenbach, and J. T. Duck (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 351).

Fig. 2, whereas the two previous specimens were surrounded by outdoor air for their entire height. It was observed that the average flue-gas temperature in the later group of chimneys was higher than that in the chimneys previously tested for the same inlet flue-gas temperature. These observations show that the gases were cooled more while traversing the chimneys in the earlier tests than for the specimens now under consideration.

The results of this series of tests on 14 chimneys of the same height, but with different masonry materials for the walls, different size and type of liners, and different treatments of the air space between the liner and the chimney wall, indicate the following conclusions regarding their performance under steady state conditions:

1. Lined masonry chimneys built of shale-tile or cinder-concrete blocks are the equivalent of lined brick chimneys in draft-producing ability and effectiveness.

2. A chimney with an enameled steel lining surrounded by 1 in. of granular insulating material produced slightly more draft than a refractory-lined chimney of the same internal diameter at low rates of gas flow (20 cfm) but no appreciable difference was apparent at higher gas-flow rates. The chimney friction was practically the same for the 7 in.-diam metal lining and the 7 in.-diam refractory lining when both were clean.

3. A 7-in. round flue lining is not the equivalent of a 7- by 7-in. square lining in capacity. The available draft increased successively in the 7- by 7-in. liners as the gas-flow rate was increased from 20 to 70 cfm, whereas it increased slightly in the 7-in. round liner for increasing flow from 20 to 45 cfm and decreased between 45 and 70 cfm. The friction loss was measurably greater in the 7-in. round flue linings at a gas-flow rate of 70 cfm than in the square liners.

4. A 10 in.-diam flue liner and a 7- by 11-in. rectangular liner having equal internal area produced about the same available draft, had about the same loss of draft due to friction, and had equal effectiveness under comparable conditions.

5. In a chimney containing two contiguous liners, one liner operating alone while the other was open at the bottom produced from 10 to 15 percent less draft than a chimney containing only one liner of the same size. When both liners received flue gases at the same temperature each produced a draft nearly equal to that produced by a chimney with only one liner.

6. Sealing off the air space in the base of a new well-constructed chimney at the bottom of the thimble had no important effect on the draft produced.

7. Grouting in the liner of a masonry chimney with mortar did not decrease the average flue-gas temperature in the chimney or the available draft produced.

These results were obtained for approximately steady state conditions under which the heat loss of the entire chimney determined the average flue-gas temperature in the chimney and hence the amount of available draft. Of equal importance is the ability of a chimney to develop a satisfactory draft quickly after hot gases initially enter the thimble, since the full-load heat output is desired from most domestic heating systems at the earliest possible moment after the heating plant is started. Under conditions of acceleration of draft, usually referred to as pickup conditions, the conductivity and heat capacity of the liner and the material in contact with it are of prime importance. It is quite possible that the specimen chimneys tested would exhibit a greater variation in performance under pickup conditions than under steady state conditions.

Although information was obtained on the several specimens for only one chimney height, the results observed for chimneys made of shale-tile and

cinder-concrete block were so nearly equivalent to those obtained for brick chimneys that the results for brick chimneys may be safely used for all practical applications.

DISCUSSION

R. K. THULMAN, Washington, D. C. (WRITTEN): This paper is an important contribution to the understanding of the functions of chimneys and their importance to the proper performance of heating equipment.

There is only one minor comment that I would like to make and that concerns the wording of Conclusion 3. From the standpoint of the home-buying public, the desirability of using a small flue of adequate capacity is apparent. Round flue linings, whether they be of tile or metal, are generally cheaper than square linings of approximately the same dimensions. Conclusion 3 compares the 7-in. round with the 7-in. square, and my objection to the wording is the unqualified statement that they are not equivalent in capacity. The tests described in the paper, and particularly the graphical presentation of the comparison of these two flues in Figs. 5, 6 and 7, show that the two flues are equivalent in capacity and that the round flue is superior in performance in terms of draft production through all the ranges of temperature and flue-gas volume up to a flue-gas volume of 70 cfm and about 550 F flue-gas temperature at the chimney inlet. It is only at capacities above this point that the performance of the round flue is not as good as that of the square. The occasions on which a chimney is required to produce a draft above 0.10 in. W.G. at temperatures above 550 F and flue-gas volumes of 70 cfm are exceedingly rare, at least as far as domestic chimneys are concerned.

It should also be noted that the areas of the two flues are not equal and that a comparison of round and square flues of equal area would show that the round flue is superior even at the higher flow rates. The comparison of the 10-in. round with the 7- by 11-in. rectangular flues, the areas of which are about the same, and which you show in Figs. 8, 9 and 10, confirms this opinion.

Many building codes specify minimum chimney sizes in terms of nominal dimensions. Often the minimum dimensions specified result in a flue area far larger than is necessary to carry off products of combustion of small house heating equipment. Codes of this sort are not only an obstacle to cost reduction in housing, but also stifle improvement in the performance of heating equipment possible with more effective chimney construction. The present wording of Conclusion 3 might conceivably be used by a code-enforcing officer to justify the continuance of antiquated dimensional expressions for minimum flue requirements.

It would be far better to revise Conclusion 3 somewhat as follows:

The 7-in. diameter round flue lining of 38.5-sq in. net area is the equivalent of a 7- by 7-in. square flue lining of 49-sq in. net area (25 percent greater) for all flue rates and temperatures up to 70 cfm flow rate and 550 F inlet temperature. Even at this unusually high flow rate and high inlet temperature, the performance of the 7-in. round in terms of observed draft was only negligibly 12 percent less than that available in the larger 7-in. square flue.

V. H. HILL, Lansing, Mich. (WRITTEN): This paper presents practical information helpful in design of combustion units which rely on chimney for venting.

One interesting point is the comparison of the draft produced by the 15-ft chimney of this paper as compared to the 15-ft chimney of the previous paper[2]. This previous paper shows an available draft range of 0.054 at 400 F entering gas to 0.094 at 1000 F. The paper under discussion shows an available draft range of 0.068 at 400 F entering

[2] Observed Performance of Some Experimental Chimneys, by R. S. Dill, P. R. Achenbach and J. T. Duck (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 351).

gas to 0.140 at 1000 F. The latter shows considerably higher drafts in spite of the cheaper chimneys used.

H. M. NOBIS, Cleveland, Ohio: I would like to see the chimney approach at an angle of 45 deg instead of at a 90-deg angle. That creates less friction, and, I think, provides the most benefit.

W. A. DANIELSON, Raleigh, Tenn.: The advantage of the metal liner, as I understand it, is for this modern plant with intermittent heating. Are data available that would clear up that point?

D. H. WYATT, Columbus, Ohio: I have heard nothing about transite flues for use in this high-temperature field. I wonder if the author could tell us if they have any material on that.

B. P. FISHER, Houston, Tex.: This discussion is mainly on flues for oil-burning equipment with liners and brick chimneys. In our area we deal mostly with gas. Our experience has shown that the aluminum liner and galvanized metal vent casing is most economical to install. It stands up well and gives good service. It should be installed according to the Underwriters' requirements, spacing it 1 in. from wood or other combustible material, as recommended by the *A.G.A.* Laboratory for "B" type vents for gas-burning appliances only.

One element many miss in venting gas is the very high dewpoint of the flue products. If you place your hand into the flue outlet of a furnace, the products of combustion will condense on your hand at body temperature. The dewpoint is about 135 to 140 deg. If you try to vent a modern gas-fired furnace into an oversized brick flue, or a terra-cotta-lined flue, you will probably have a great deal of condensation at the beginning, because it is hard to preheat that cold flue or chimney. That is why the aluminum-lined flue does a better job of venting the products of combustion from gas. It heats quickly and allows that water vapor to be carried out, and consequently creates a draft. I have found that, in some cases where an oversized flue is used, it is beneficial to install an opening 12 to 14 in. below the furnace-vent connection to the flue. This opening should be equal in area to the difference in area between the furnace-vent collar and the inside lined area of the flue. This arrangement allows the cold air column to fall down and the light, warmer flue products, wet with water vapor, to be carried out.

The venting of gas products of the modern gas-burning appliance is a study in itself. I would recommend to all a paper on Gas Appliance Venting by Arthur Theobald of the Payne Furnace Co.

MR. THULMAN: For the benefit of those who might conclude that this paper is a report of a complete investigation at the Bureau, it is actually only a part of what originally was planned as a comprehensive chimney research program. Reference was made in the paper to the previous tests which were run in 1942 and the data resulting from those tests are now in THE GUIDE. Prior to publication of these data in THE GUIDE, there was very little information about chimneys for the small, residential-type installation. The program as originally contemplated embraced three temperature ranges, the first of which would show the operating performance of the chimney in terms of draft-producing ability at the range of temperatures normally encountered in domestic heating appliances, and that range from 200 to 1000 F is the range that Mr. Achenbach has reported today.

The codes governing the various heating appliances that are used in residential work permit flue-gas temperatures at the outlet for certain appliances of 1000 deg. Mechanically-fired boilers are limited to 600; gas appliances, 550. Actually, with the requirement of a draft diverter in the gas appliance, the flue-gas temperatures entering the chimney are somewhat less than 550, and sometimes as low as 200.

Another range of temperatures is contemplated covering the question which Mr. Fisher raised, and that is, the extremely low temperatures. This range would overlap the others, and we contemplate something like 90 deg up to about 250 or 300 for this lower range.

A third phase of the program involves the investigation of the safety of chimneys in terms of temperatures in excess of 1000 deg and ranging from about 800 up to 1800. These tests have been conducted by the Fire Protection Section of the *Bureau of Standards* on a number of similar masonry chimneys. This, again, is only one part of a comprehensive program.

L. B. SCHMITT[3] and R. B. ENGDAHL, Columbus, Ohio (WRITTEN): For a number of years there has been a need for definite information on the operational characteristics of various types of small residential chimneys, and it is, indeed, fortunate that at this time we should have such a complete answer to this problem.

Our work on residential chimneys conducted at *Battelle Memorial Institute*, and recently presented in the paper entitled, Performance of Residential Chimneys, parallels the work presented by Mr. Achenbach and Mr. Cole in many respects. It is gratifying to know that our results are so closely comparable with the data that they have presented.

Chimney effectiveness as presented in Table 2 should be clarified more fully. Chimney effectiveness, or chimney efficiency as we have called it, is the ratio of the observed draft to the computed static draft calculated from the inlet temperature. The computed static draft as presented in the paper is calculated from the average flue-gas temperature, and is used to determine the indicated friction loss. This is a technicality that may easily cause confusion.

It will also be noted in Table 2 that the observed draft exceeded the computed draft at a number of points. A draft difference of 0.004 in. of water is equivalent to only 11 deg difference in temperature. Mr. Achenbach has stated that unshielded, chromel-alumel thermocouples were used, and, with thermocouples of this type, this degree of error is not unusual. No such difference was noted among the measurements that we recorded, which were made with a high-velocity aspirating shielded thermocouple.

AUTHORS' CLOSURE (Mr. Achenbach): I wish to thank those who have made comments on this paper. Mr. Thulman's written comment about Conclusion 3 is a good criticism. The statement as made is not inaccurate, but probably is not sufficiently complete. Since the comparison of capacity between round and square liners is a controversial one, the conclusions should be somewhat more detailed. I believe the text of the paper covers the point adequately.

A smoke pipe that entered a chimney at an angle of 45 deg would probably introduce less friction and possibly provide a better streamlined flow of the gases in the chimney than one entering at right angles. The use of a thimble inclined upward at a 45 deg angle has not been widespread in my experience, however. The Bureau has made some tests to compare the pickup of a number of these chimneys and there is an appreciable difference. These data have not all been correlated as yet, but they will be published later. The scope of the investigation reported in this paper was intended to cover the flue-gas temperatures that might occur with gas appliances as well as with oil and coal and was not related to one fuel in particular.

I believe that the use of transite liners is limited almost entirely to gas appliances, since the transite liner tends to disintegrate at a temperature a little above 600 F. The use of aluminum liners may also be limited to gas. The point is somewhat controversial, since aluminum melts at about 1220 F. Tests of one such chimney at the *National Bureau of Standards* showed that an aluminum liner might melt under conditions that occur occasionally with oil burners, and more often with solid fuel.

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[3] Research Engineer, *Battelle Memorial Institute.*

Chimney effectiveness, as used in this paper, and chimney efficiency, as used by *Battelle Memorial Institute,* are the same quantity. It depends only on the observed draft and the inlet flue gas temperature for any given test condition. The computed static draft, discussed in this paper, is the draft that would be observed if the flue gases passed through the chimney without friction, but were normally cooled during passage. The computed static draft was determined from the average flue gas temperature in the chimney and was used to determine an indicated friction loss. In this way, the error in temperature measurement caused by using unshielded thermocouples is reflected in the values obtained for the indicated friction loss. The radiation error of the unshielded thermocouples reduced the value of the indicated friction loss, and, in some cases, caused it to have a negative value as explained in the text of the paper. Computations indicate that the radiation error of the unshielded thermocouples caused an error in the computed static draft ranging from 0 to 2 percent for the various test conditions.

Although this paper indicates that the chimneys made of different masonry materials were comparable and, for all practical purposes, equivalent in their ability to produce a draft, they were not necessarily equivalent from a fire hazard standpoint. I am not prepared to comment in detail on the fire hazard characteristics of this group of chimneys. However, all of these that were discussed in this paper, and some additional ones, have been tested to determine whether they would ignite surrounding combustible material, and also whether the masonry would crack due to heat shock. These tests have shown that only one of them would ignite combustible material for flue gas temperatures less than 1000 F. However, when flue gases at a temperature of 1400 F were admitted for one hour, followed by a steady flow of flue gases at 1000 deg, I believe all but about two did ignite combustible material, or had reached temperatures where ignition was imminent.

1357

No. 1357

PERFORMANCE OF AN ELECTRICAL SYSTEM OF PANEL HEATING WITH FOUR STAGES OF INSULATION

By R. J. Lorenzi*, New Haven, Conn. and J. F. Schreiber**, Passaic, N. J.

NUMEROUS articles have appeared in behalf of the economic and comfort advantages of insulation when used in conjunction with heating equipment. The authors, however, are not familiar with any publication in which a cost analysis has been made based on actual experimental data taken in a full-scale residence heated electrically and tested with varying degrees of insulation.

This article deals with the performance and economy of operation of a panel heating system in which the source of energy is a layer of electrically conductive rubber. The installation was made in a full-scale experimental four-room house on the grounds of the John B. Pierce Foundation at Raritan, N. J. Data which were obtained during the 1947-1948 heating season are presented for four progressive stages of insulation.

During the entire heating season, the test house was occupied by an elderly couple and their daughter. All normal household operation, such as cooking, washing, etc., were carried on with no restrictions. The occupants were permitted to set the room thermostats to satisfy their own comfort requirements, except during 24-hour periods of continuous temperature recording, which occurred approximately once every two weeks. The occupants were asked to note the thermostat settings.

Description of Test House

Fig. 1 shows an exterior view of the test house which is built of the prefabricated sandwich type wall construction. The wall panels are composed of 2 in. mineral wool insulation compressed to 1½ in. between sheets of ⅜ in. plywood.

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** Development Engineer, Mechanical Goods Division, United States Rubber Co.

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No changes were made in the wall construction during the various experimental phases of the program. Windows are of the casement type and were of single glass for the first insulation stage, but were fitted with double glazing for the second, third, and fourth insulation stages. The original ceiling was 3/8 in. plywood panels above which was 2 in. mineral wool insulation, an attic space and a roof constructed of asbestos shingles on 1/2 in. plywood sheathing. For the series of experiments described in this paper, the original ceiling was furred down using 2 in. x 2 in. furring strips, and the heating panels were nailed to these strips, thus providing an additional air space. After the second insulation step, an additional 2 in. thickness of mineral wool insulation was added in

FIG. 1. FOUR ROOM RESIDENCE USED IN TEST

the ceiling, thereby providing a total of 4 in. of insulation in Steps 3 and 4. Since the test house had been used for experiments other than those described herein, the floor construction is somewhat unusual, and is as follows: Living room, 1 in. finished flooring over 5/8 in. plywood sub-flooring, 2 in. fiber glass on backer board tacked to bottom of floor joists; Dining-kitchen space, 1 in. finished flooring over 5/8 in. plywood sub-flooring, backer board tacked to bottom of floor joists; Bathroom and Bedroom No. 1, 1 in. finished flooring over reflective insulation draped over floor joists; Bedroom No. 2, 1 in. finished flooring over reflective insulation draped over floor joists, backer board attached to bottom of floor joists. For the final step in insulation, an additional 2 in. thickness of mineral wool was added to the floor construction in all rooms except the living room.

The overall coefficients of heat transfer for the various surfaces subject to heat loss are given in Table 1 and are expressed in Btu per square foot per hour per Fahrenheit degree. Values are given for each insulation step, and the underlined figures indicate where insulation changes occurred. Heat loss data were obtained on the basis of the generally accepted procedure for heat loss calculations as presented in the HEATING VENTILATING AIR CONDITIONING

GUIDE. Infiltration was obtained by the crack method, assuming the average wind velocity to be 15 mph during December, January and February. The heat loss for the structure for the first insulation step was 34,000 Btu per hour. This does not include the ceiling loss which was taken into account by the reverse loss from the heating panels.

DESCRIPTION OF HEATING SYSTEM

The radiant panel ceiling heating system was of a type which consists of a natural rubber to which has been added specific materials permitting the conduction of electricity. The rubber provides a uniform heating surface over the

TABLE 1. COEFFICIENTS OF HEAT TRANSMISSION FOR SURFACES SUBJECT TO HEAT LOSS[a]

(Btu per square foot per hour per Fahrenheit degree)

	STEP 1	STEP 2	STEP 3	STEP 4
Walls—Lintel Girder Panels	0.21	0.21	0.21	0.21
Walls—Middle and Lower Course Panels	0.18	0.18	0.18	0.18
Windows	1.13	0.55	0.55	0.55
Ceiling and Roof (overall coefficient)	0.10	0.10	0.06	0.06
Floor—Living Room	0.08	0.08	0.08	0.08
Floor—Dining Room-Kitchen	0.21	0.21	0.21	0.08
Floor—Bedroom No. 1 and Bath	0.27	0.27	0.27	0.09
Floor—Bedroom No. 2	0.24	0.24	0.24	0.09

[a] Underlined figures show where insulation changes occurred.
Wall, window, and roof coefficients based on a wind velocity of 15 mph.

entire area. The conductive rubber layer is sealed between layers of phenolic impregnated insulation and the laminated construction is made rigid by a backing of 3/16 in. asbestos board. The completed bonded assembly is approximately 1/4 in. in thickness. Three geometrical sizes of heating panels were used for the installation—3 ft x 3 ft, 3 ft x 4 ft, and 4 ft x 4 ft. The panel layout for each room is indicated in Fig. 2, which is a plan of the test house giving room dimensions. Two electrical densities were utilized—12 and 16 w (watts) per square foot, the higher density panels being located near the walls subject to exposure. Panel numbers 6-9, 12-14, 20, and 21, are all rated 12 w per square foot, and the remaining panels are rated at 16 w per square foot, with the exception of No. 11, which is a filler panel used to complete the geometrical pattern and to meet the architectural requirements. It will be noticed that in some rooms (bathroom, bedroom No. 1, and dining-kitchen area) it was necessary to use 16 w density panels throughout to satisfy the heat loss requirements. Wherever possible, full ceiling coverage was used to reduce the panel temperatures to a minimum value.

All panels were operated on 220 volts, and the electrical connections to the panels were made with standard building wire of the armored cable type attached through connection boxes fastened to the backs of the panels. There were four main circuits and the total installed capacity was 7300 w. The system

was controlled by means of separate thermostats in each room. These thermostats were of the standard low-voltage air actuated type with operating differentials not in excess of ±0.5 deg.

ELECTRICAL AND THERMAL MEASUREMENTS

Since normal occupancy was considered to be a critical item in these tests, the physical nature of the recording equipment, its location, and psychological acceptability by the occupying family provided a number of problems. The

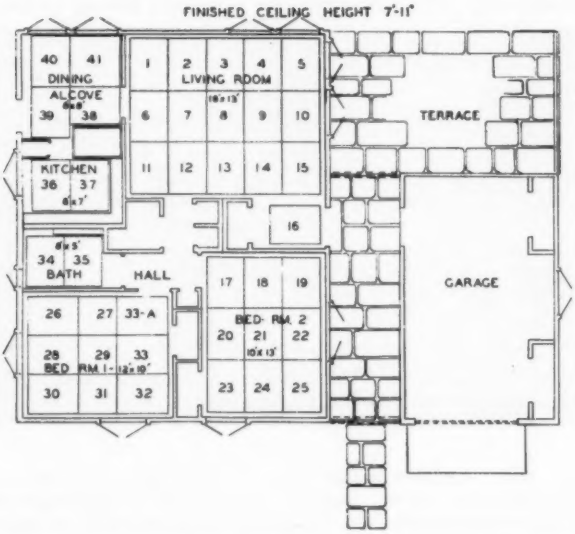

FIG. 2. PLAN OF TEST HOUSE SHOWING HEATING PANEL LAYOUT

arrangements finally decided upon were believed to meet the full requirements of informative electrical and thermal performance records with a minimum of inconvenience to the family and without conspicuous external equipment in the living spaces. Electrical data were obtained on total energy consumption, momentary demand, and consumption of power in each controlled area by the use of a watt-hour meter, a recording watt meter, and time meters connected to each area tested. From the data obtained from the instruments, it was possible to determine the system's power consumption, the total load (demand) at any given time, the diversity of this demand with respect to time, and the proportion of energy consumed by any one area for a given length of time.

Thermal measurements were obtained primarily through the use of copper-constantan thermocouples in conjunction with a continuous type chart recorder. In each of the four major rooms on the approximate centerline of one exterior

and one interior wall, and 6 in. from these walls, air thermocouples were located 30 in. above the floor. Surface temperature measurements were obtained at the ceilings in all rooms, and thermocouples were also located inside the zone thermostats in each room. To obtain outside air temperatures, both thermocouples and thermometers were used, and wind velocity was obtained by the use of an integrating anemometer.

EXPERIMENTAL DATA

A summary of the pertinent experimental data is presented in Table 2. The temperatures of lines 1 and 2 of this table are averages of recordings taken at 15-min intervals for a 24-hr period for each step, and are averages of the four major rooms. The temperatures of line 2 were measured at the lower surface

TABLE 2. SUMMARY OF PERTINENT EXPERIMENTAL DATA

	UNIT	STEP 1	STEP 2	STEP 3	STEP 4
1. Air Temperature at 30 in. level	F	71.7	72.5	71.7	71.2
2. Ceiling Panel Temperature	F	100.8	99.0	106.2	103.2
3. Indoor-Outdoor Temperature Difference	F deg	45.1	49.9	54.7	56.8
4. Average Wind Velocity	mph	5.8	5.0	7.9	4.6
5. Total Duration of Test Condition	hr	672	504	168	168
6. Total Degree-Days		901	819	321	334
7. Energy Consumption	kwhr	4101	2680	990	964
8. Unit Power Requirement	kw/deg	0.136	0.107	0.108	0.101
9. Heat Factor[a]	kwhr/degree-day/1000 cu ft	0.70	0.51	0.48	0.45

[a] Heat Factor Formula to Calculate Electric House Heating, by H. C. Bender, *Electrical World*, May 12, 1945. This *heat factor* is a common denominator by which comparisons can be made and by which energy consumption may be estimated for houses of different size and construction under varying climatic conditions. An analysis of available national information showed that the *heat factor* varied from 0.8 to 0.2 and that similar types of houses had similar heat factors.

of the heating panels. For lines 3 and 4, the data presented are for the entire test period, that is, for the number of hours shown in line 5. Line 6 is the total degree days for each step as calculated from a 65 F base. Line 7 is the total energy consumed for each test period for house heating only. Line 8 is the total unit power requirement per degree of inside-outside temperature difference, and is line 7 divided by the product of lines 5 and 3. The heat factor of line 9 is another expression of the unit power requirement and is given in terms of kwhr per degree-day per 1000 cu ft of heated volume. Note that this expression uses the degree-day concept and that, in calculating line 8, the inside-outside temperature difference was used. Both lines 8 and 9 indicate the measured reduction in unit power requirements for the successive insulation steps, but the heat factor formula of line 9 shows this reduction more positively.

In line 1 of Table 2, the temperatures presented may appear at first glance to be somewhat high for a radiant panel heating installation. However, when it is recalled that the occupants of the test house were an elderly couple, the temperatures appear to be more reasonable. The ceiling panel temperatures of line 2 also are of interest. Note that the highest panel temperature is 106.2 F, occurring with an indoor-outdoor differential of 54.7 deg and that for Step 4 (the condition of greatest insulation) the ceiling panel temperature is only

103.2 F for a slightly greater indoor-outdoor temperature differential. In the design of the heating installation, the temperature produced by a 16-w density panel, approximately 110 F at 0 F outdoor, was selected as a maximum panel temperature for optimum comfort for the 4th insulation step.

From line 9 of Table 2, the reduction in heat factor from 0.70 to 0.51 indicates a saving of approximately 27 percent due to the addition of the storm windows. This checks well with a theoretical saving of 26 percent obtained by computing the heat loss of the experimental house with and without storm windows. However, this saving is greater than normally expected of storm windows and is due to the relatively large glass area in the small house, and the reduction in infiltration loss when the storm windows were added. (The casement windows were not weather stripped.)

COST DATA

The summary of the insulating and operating costs is presented in Table 3. In this table, line 1 is the heat factor which is the same as line 9 of Table 2. In line 2 of Table 3, the cost of the heat loss per heating season is presented. These data are based on the heat factor determined by experiment and are the products of the heat factor, the degree-days—5586 for Somerville, N. J.—times 6.48 which is thousands of cubic feet of heated volume, times 2 cents per kwhr plus $24.00 per year demand charge. Electrical energy was purchased from the local utility at the rate of 2 cents per kwhr for all amounts in excess of 200 kwhr per month. There was also a demand charge of $2.00 per kw in excess of 7.5 kw or $24.00 per year.

In line 3, the cost of the insulation investment is given. Under Step 2, $183.00 represents the installed cost of the storm sash. For Step 3, $254.70 represents the installed cost of the storm sash plus the additional ceiling insulation. In Step 4, $341.30 is the combined cost of storm sash, ceiling insulation, and additional under-floor insulation. These cost figures are not estimated but are the actual installed cost figures. The amortization cost of line 4 is given in dollars per dollar invested per year, and is based on 20 years at 5 percent. The annual insulation cost of line 5 is obviously the product of line 3 and line 4, and the combined annual costs of line 6 are the totals of lines 2 and 5. Lines 7 through 10 show the savings during and after the amortization period, both in dollars and in percent. It is obvious from line 6 of Table 3 that each successive insulation step has been justified economically, since the combined cost of heating plus insulation investment is progressively reduced with the addition of more insulation.

In line 2 of Table 3, the calculation was based on the actual cost of electrical energy prevailing at Raritan, N. J. The rate in that locality is such that any type of electrical heating falls into a class which might be termed *luxury heating*. Since more logical applications for electrical heating would be in areas where energy could be purchased for approximately 1 cent per kwhr, the annual heating costs have been calculated on this latter basis in the center section of Table 3. It has also been assumed that the test house was subject to the same exposure and heat loss as in tests reported, and that the demand charges and insulation costs are the same, but that the average energy cost is 1 cent per kwhr. The cost of the heat loss on this basis is given in line 11, and the combined annual costs are given in line 12. With this reduced rate, it can be seen that the

successive insulation stages may still be justified economically, since the combined cost for Step 4 gives the lowest value.

In the last section of Table 3, a further reduction to an average rate of ½ cent per kwhr has been assumed, and the calculated values are given in lines 13 and 14. With this low rate, notice that the final insulation step cannot be justified, since the combined annual cost exceeds that for Step 3. An average rate slightly above ½ cent per kwhr is, therefore, the lowest rate at which all stages of insulation can be justified in this particular instance.

COMFORT OBSERVATIONS

During the entire heating season operation continued without difficulty, both panels and controls functioning properly. Good response to the *On-Off* type

TABLE 3. SUMMARY OF COST DATA

	STEP 1	STEP 2	STEP 3	STEP 4
1. Heat Factor, kwhr per degree-day per 1000 cu ft	0.70	0.51	0.48	0.45
2. Heat Loss Cost per Season, dollars at 2 cents per kwhr	530.61	395.94	370.29	346.64
3. Net Insulation Investment, dollars	——	183.00	254.70	341.30
4. Amortization Cost, dollars per dollar per year	——	0.08	0.08	0.08
5. Annual Insulation Cost, dollars	——	14.64	20.38	27.30
6. Combined Annual Cost, dollars	530.61	410.58	390.67	373.94
7. Annual Saving During Amortization, dollars	——	120.03	139.94	156.67
8. Annual Saving After Amortization, dollars	——	134.67	160.32	183.97
9. Percent Reduction in Annual Costs During Amortization	——	22.6	26.4	29.5
10. Percent Reduction in Annual Costs After Amortization	——	25.4	30.2	34.7
11. Heat Cost per Season, dollars at 1 cent per kwhr	277.30	209.97	197.15	185.32
12. Combined Annual Cost, dollars	277.30	224.61	217.53	212.62
13. Heat Loss Cost per Season, dollars at ½ cent per kwhr	150.65	116.99	110.58	104.66
14. Combined Annual Cost, dollars	150.65	131.63	130.96	131.96

room thermostat was due to the minimal heat capacity and low thermal lag of the heating panels. This made it possible to control the panel heating system without an outdoor thermostat. In general, the occupants reported comfort during the test periods in each insulation step with the exception of Step 1 in which no storm sash was used. An unsolicited statement by the occupant indicated considerable improvement in comfort by the addition of the storm sash. Comments on cold floor conditions in some rooms (which were expected due to the unusual floor construction of the test house—see description of floor construction given previously) resulted in the addition of the under-floor insulation in Step 4 and showed the necessity of the insulation for this type of construction.

SUMMARY

1. A full-scale residential panel heating installation, using electricity as the source of heat, has been described.

2. Heating panels fabricated of electrically conductive rubber and controlled by *On-Off* air actuated thermostats were utilized.

3. Comparative temperature and power data have been presented for four progressive stages of insulation.

4. Cost data have been presented to show that the successive insulation stages can be justified economically under certain conditions of power cost and climatic severity.

5. Performance was good throughout the heating season, and comfort was reported by the occupants for the final insulation stages.

ACKNOWLEDGMENT

The authors wish to thank the U. S. Rubber Co., which supplied the heating panels and sponsored the tests, and the following members of the Pierce Foundation for their helpful advice and assistance: G. M. Rapp, assistant executive director, Dr. L. P. Herrington, director of research, Physiological Research Division, John Cataldo and E. deF. Tidd, Electrical Division, W. F. Koppes, Architectural Division, E. J. Zelezny, Thermal Division, and also the members of the U. S. Rubber Co. for their helpful suggestions: Dr. B. W. Bender, Mechanical Goods Division, and R. C. Cassidy, Electrical Wire and Cable Department.

DISCUSSION

H. C. MURPHY, Carmel, Calif. (WRITTEN): We have approximately 800 installations of electrical panel heating on the Pacific Coast and in certain other sections of the country having power rates of approximately one cent per kilowatt hour.

Some of these systems have been in use up to eight years. The two different, but somewhat basically similar systems are manufactured in Seattle, Wash., and in Monterey, Calif.

Inspection of a number of these jobs confirms the major importance of adequate insulation. With proper insulation, adequate controls, and good engineering, this method of heating seems to be practical and desirable for many types of structure.

E. P. PALMATIER, Syracuse, N. Y.: Will the authors please explain the basis of calculation of the factor 0.5 kw per degree day per thousand cubic feet which is given in the tabular data. I have seen lower figures than this quoted, but it may be that the volume of the houses was not computed by the same method used by the authors. Exactly how was the house volume determined?

R. A. MILLER, Pittsburgh, Pa.: Do the authors mean *double glazing* or *storm sash?* The factor given indicates double glazing. The words used were storm sash. There is some difference in forming calculations and making a determination.

M. C. CHRISTESEN, Newark, N. J.: Were any records made of the differences in temperature between the ceiling and the floor during the operation of the panel. What were those differences for each foot of the height from floor to ceiling?

H. M. HART, Chicago, Ill.: Lack of discussion is a compliment to the authors. They have written a paper which leaves little for discussion.

C. H. RANDOLPH, Milwaukee, Wis.: Did the authors find anything in their work that would indicate that one would figure fuel costs for an electric panel heating system on any basis other than the one used for an ordinary standard conventional system?

A. S. CHIPLEY, Libertyville, Ill.: Was the panel control an on-and-off proposition or was it possible to modulate the amount of electricity through the panels?

CYRIL TASKER, Cleveland, Ohio: In view of the costs given for net installation investment and because, in different parts of the country, such costs are liable to differ somewhat, is there any statistical significance in the figures for Steps 2, 3, and 4 when the cost of electric power is more than one cent per kilowatt hour? I would suggest, on the basis of the figures given, that actually no step beyond Step 2 or the first step of insulation is really significant. As can be seen from the figures, some slight change in the net installation investment might easily wipe out the small differences between Steps 2, 3, and 4 at one cent per kilowatt hour or below.

R. E. REEVES*, Chattanooga, Tenn.: We have two or three installations of this type in Chattanooga and also of the type being used on the Pacific coast. We have approximately 56,000 kw of electric house heating in Chattanooga, representing some 4,000 all-electric homes. We find that the houses in which the radiant panels have been installed are more comfortable than those where straight resistance heating is installed. Owners find (our rates being somewhat lower in the Tennessee Valley area) that the electrical cost is considerably below what the authors have presented. There are many houses that heat for less than $100 per season, averaging 7½ mills per kilowatt hour.

M. K. FAHNESTOCK, Urbana, Ill.: The authors mentioned comfort. Did they have any objective measurement of comfort? How did they evaluate comfort? Did they just take the reactions of the subjects? How many subjects were questioned? Was there anything in regard to the state of equilibrium of the subjects? In other words, how did they define comfort and how did they measure it?

R. L. REILEY, Texarkana, Ark.: Is the rubber material mentioned by the authors an established material or is it an experimental one? From whom can it be obtained?

AUTHORS' CLOSURE (Mr. Lorenzi): I thank Mr. Murphy for his written discussion, and Mr. Hart for his comments.

Mr. Palmatier wants to know about the heat factor: What volume of the house was used? The volume was the entire volume of the occupied area, including the partition volume. That is, taking the entire floor area as based on the full length and width of the house and multiplying by the height from the bottom of the floor right up to the ceiling. This volume was calculated according to the recommendations of H. C. Bender, to whom we should acknowledge the use of the heat-factor formula.

Mr. Miller wanted to know about storm sash. We did use storm sash and not double glazing. Mr. Christesen asked about a differential gradient from ceiling to floor, or floor to ceiling. I have no specific figures but estimate that the floor-to-ceiling grade was of the order of 10 to 12 F as measured from a point six in. above the floor to a point six in. below the ceiling. We found in most cases that the actual surface of the floor itself was warmer than the air immediately above it.

As to the question on the type of controls, these were of the on-off type. There was a thermostat located in each of the four major rooms and also one in the bathroom. We actually had five zones of control, each one operating independently. Each one was of the air-actuated type.

Mr. Randolph asked about figuring fuel for electrical heating; whether it was necessary to figure fuel cost for electrical heating on a basis other than that used for regular standard types of heating. Our electrical demand charts showed that the electrical demand followed very closely the outside temperature. I think that is an indication that the use of the present day concept of predicting fuel costs is a pretty reliable method.

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* Member, Electric Power Board of Chattanooga.

Mr. Tasker's question was on the economic significance of Steps 3 and 4. I believe Mr. Tasker is correct. There is no considerable saving in the amount of fuel by the addition of an extra two in. of insulation in the ceiling when we already had two in. there. This is also true for the floor. However, in the case of the floor the comfort conditions were improved tremendously by adding the extra insulation. That was indicated not only by the reactions of the occupants, but also by the thermocouple measurements taken at the floor surface.

Mr. Reeves has made some comment about the more favorable cost of electrical heating in the Chattanooga section. Of course, the degree days there are much more favorable than they were in the New Jersey location, where we did the testing. I think that the Chattanooga area in the Tennesse Valley, is a much more likely location for the use of electrical heating.

Prof. Fahnestock was interested in knowing about how comfort was defined and how it was measured. We made no attempt to obtain elaborate information on comfort from occupants or to measure it by having reactions from subjects. The house was occupied during the entire test period by three adult persons and our only information was the information which they offered to us.

Mr. Reiley asked about the rubber materials used. This rubber is made electrically conductive by addition of carbon black. By varying the amount of carbon black which is added to the material, the electrical watt density can be varied. The material is available commercially in the form of panels in two sizes and two watt densities.

1358

No. 1358

HEATING A BASEMENTLESS HOUSE WITH RADIANT BASEBOARD

By R. H. WEIGEL* AND W. S. HARRIS**, URBANA, ILL.

DURING the past year, the University of Illinois, in cooperation with the *Institute of Boiler and Radiator Manufacturers,* investigated the performance of a radiant baseboard heating system in a basementless home. This supplements previous work done on radiant baseboard in the I=B=R Research Home[1, 2].

The object of the tests discussed in this paper was to determine the operating characteristics of a hot-water heating system using type RC, cast-iron, radiant baseboards in a basementless home. These characteristics included the temperature of the floor slab, the temperature gradients in the rooms, the operating temperatures in the hot-water system, fuel consumption, relative humidity, and cleanliness of operation.

DESCRIPTION OF EQUIPMENT

House: The basementless home, shown in Fig. 1, was of prefabricated construction. The walls were composed of ⅜ in. fiber wall board on the room side, 2 in. by 3 in. wood framing members, and ⅜ in. plywood on the outside. One inch of cotton insulation, provided with a vapor barrier, was applied to the wall board between the framing members. The calculated coefficient of heat transmission, U, for this wall section was 0.16 Btu per (sq ft) (hr) (F deg) temperature difference. The ceiling was composed of ¼ in. plywood, two inches of cotton insulation, and 2 in. by 4 in. joists. The U value was 0.13 Btu per (sq ft) (hr) (F deg) temperature difference. The house was erected on a 4-in.

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[1] A Study of Radiant Baseboard Heating in the I=B=R Research Home, by Alonzo P. Kratz and Warren S. Harris. (University of Illinois, *Engineering Experiment Station, Bulletin* 358.)

[2] Performance of Radiant Baseboard in Research Home, by A. P. Kratz and W. S. Harris. (*Heating, Piping & Air Conditioning,* Vol. 18, No. 1, Jan. 1946, p. 138.)

Presented at the 55th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January, 1949.

FIG. 1. BASEMENTLESS HOME

concrete floor, and, to minimize the heat losses at the edge of the slab, 1-in. waterproof board insulation was installed around the periphery and between the slab and foundation. A vapor barrier was placed between the slab and gravel fill to prevent vapor transmission. Calculated heat losses through the floor were based on a factor of 0.06 Btu per (sq ft) (hr) (F deg) and a temperature difference from room air to ground of 20 F deg.

All windows and doors were weatherstripped, and storm doors were used on both outside entrances. The total heat loss, based on —10 F outdoor temperature, and 70 F indoor temperature, was 42,750 Btu per hr. The total floor area of the heated space was 713 sq ft and the volume was 5702 cu ft. Table 1 gives a summary of the calculated heat loss, the volume, and the installed radiation in each room in the house. A floor plan of the house is shown in Fig. 2, and construction details of floor and walls are shown in Figs. 3 and 4.

TABLE 1. DATA ON BASEMENTLESS HOME AND HEATING SYSTEM

ROOM	DIMENSIONS	HEATED SPACE CU FT	CALCULATED HEAT LOSS BTU PER HR	INSTALLED RADIATION LOW HEIGHT TYPE RC	
				Lineal Ft of Radiant Baseboard[a]	Radiation Installed Sq Ft[b]
Utility Room	8 ft-0 in. x 11 ft-10 in.	755	5928	—	—
Kitchen	9 ft-3 in. x 11 ft-10 in.	784	6384	10	20.8
Living Room	11 ft-10 in. x 19 ft-10 in.	1869	13680	26	54.1
Front Bedroom	9 ft-11 in. x 11 ft-10 in.	935	7182	15	31.2
Rear Bedroom	8 ft-9 in. x 11 ft-10 in.	831	6612	14	29.2
Bath	5 ft-0 in. x 8 ft-1 in.	324	2394	5	10.4
Hall	7 ft-1 in. x 3 ft-7 in.	204	570	—	—
	Total	5702	42750		145.7

Net I=B=R Boiler Rating = 55000 Btu per hour

Gas Burning Rate = 100 cu ft per hour

[a] Based on a heat emission of 500 Btu per lineal ft at an average water temperature of 215 F.
[b] Based on a heat emission of 240 Btu per hour per sq ft at 215 F.

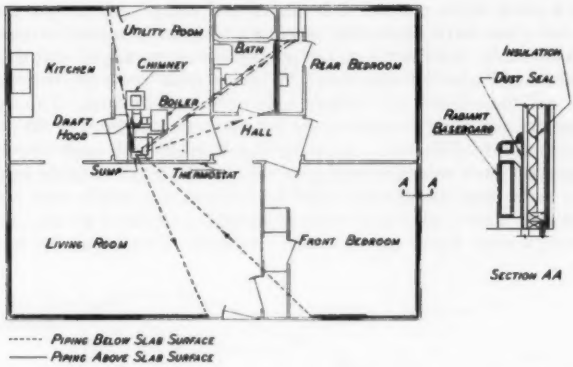

FIG. 2. FLOOR PLAN SHOWING LOCATION OF BASEBOARD RADIATION

Heating System: A gas conversion burner was used in conjunction with a wet-base type cast-iron boiler composed of two 6-in. sections and one 4-in. section, surrounded on top, sides and back with a 1-in. thickness of air-cell insulation and completely enclosed in an enamelled sheet-metal jacket. All cracks between sections were sealed with asbestos cement. The net I=B=R rating was 55,000 Btu per hr.

A one-pipe, forced-circulation, hot-water system was used with type RC, cast-iron, radiant baseboards. These units, approximately 7 in. high, were located in the rooms as shown in Fig. 2. Paper-backed aluminum foil served as a dust seal to prevent the possibility of streaking the wall. The radiation in the living

TABLE 2. RADIATION, PIPE, AND FITTINGS USED IN HEATING SYSTEM WITH RADIATION CONNECTED IN TWO CIRCUITS

SECTION	INSTALLED RADIATION	PIPE		ELBOWS		TEES		UNIONS		SPECIALTIES	
	Low Height Type RC Baseboard Radiation, Lineal Ft	Size, In.	Length, Ft	Size, In.	No.	Size, In.	No.	Size, In.	No.	Unit	No.
Bath-Bedroom Circuit	34	¾	54.0	¾ ¾-union	10 4	¾	1	¾	2	¾-in. valves	2
Liv. Rm.-Kitchen Circuit	36	¾	36.2	¾ ¾-union	8 4	¾	2	¾	2	¾-in. valves	2
Supply Trunk		½ ¾	2.9[a] 14.3[b]	½ ¾	3 2	¾x½x¾ ¾	1 1	½ ¾	1 2	¾ relief valve	1
Return Trunk		1	0.5	1	1	¾x1x¾ 1x1x½	1 1	1	2	1-in. circulator	1

Pipe Totals	*Elbow Totals*	*Tee Totals*	*Union Totals*
½ in.....2.9 ft	½ in.....3	¾ in.....4	½ in.....1
¾ in.....104.5 ft	¾ in.....20	¾x½x¾ in....1	¾ in.....7
1 in.....0.5 ft	¾ in. (union)..8	¾x1x¾ in.....1	1 in.....2
	1 in.....1	1x1x½ in.....1	

[a] To expansion tank
[b] 6 ft drain from relief valve

room and kitchen were connected in series and comprised one circuit, the two bedroom radiators were connected in series to form a second circuit, and the bath a third circuit. Furthermore, the piping was so arranged that the radiation in the bath could also be connected in series with the radiation in the two bedrooms and thus divide the system into only two circuits. The system was designed to operate with a mean water temperature of 215 F at 80 deg indoor-outdoor temperature difference, and a 20 deg drop through each circuit.

Three-quarter inch wrought-steel pipe was used almost entirely in the piping layout. In order that the system could be drained, the supply and return pipes in the concrete slab (see Fig. 2) were pitched downward to the sump in the utility room where drain valves were provided. Three-quarter inch air-cell

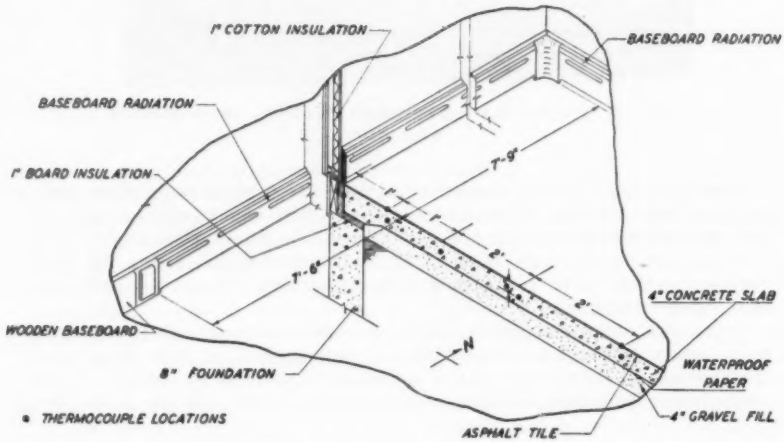

FIG. 3. THERMOCOUPLE LOCATIONS IN LIVING ROOM FLOOR SLAB

insulation was placed around the pipe buried in the concrete. The insulation permitted sufficient movement of the pipe to provide for expansion in the system due to temperature changes and also it made it easier to connect the radiation to the supply and return piping at the time of installation. Each baseboard assembly was provided with an air vent to eliminate any air that might accumulate in the radiation or piping. The amount of radiation, together with the size and amount of pipe and fittings, are given in Table 2.

Control System: The control system, shown in Fig. 5, included a room thermostat, a high-limit control, a magnetic gas valve, a safety pilot, a transformer, and a relay. The room thermostat was of the heat-anticipating type, and in addition, contained provision for making three adjustments of the length of the burner and circulator operating cycle for a given outdoor temperature. The thermostat was located 30 in. above the floor on the east wall of the living room near the hall entrance, see Fig. 2. The high-limit control, which served to prevent overheating of the water in the boiler, was of the immersion type and was located in the back section of the boiler.

When heat was required, the gas burner and the circulator were started by the thermostat and both continued to operate until stopped by the room thermostat. If the period of operation was of sufficient duration to raise the temperature of the water in the boiler to 220 F, corresponding to the setting of the high-limit control, the latter would stop the burner. The circulator, however, would continue to operate. This system of control was used since domestic hot water was not being provided by the boiler.

Testing Apparatus: While the house was being erected, copper-constantan thermocouples were located in the living room and kitchen floor slabs, as shown in Figs. 3 and 4, respectively. It may be noted that slab temperatures in the living room were measured in a plane perpendicular to the west wall and that

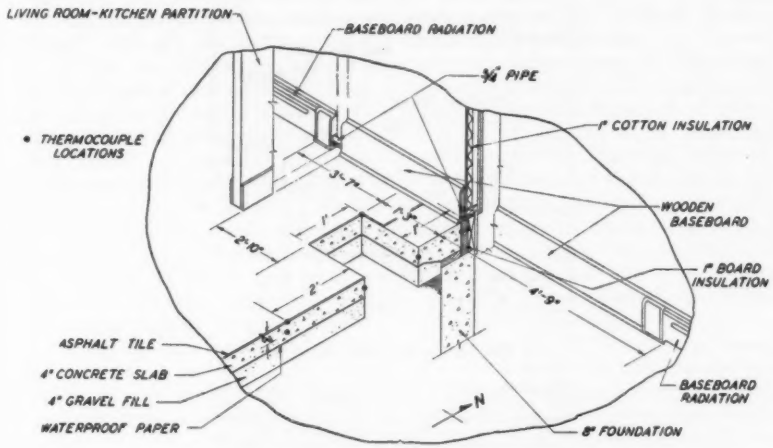

FIG. 4. THERMOCOUPLE LOCATIONS IN KITCHEN FLOOR SLAB

baseboard radiation extended not less than 7 ft-6 in. on either side of the plane in which the thermocouples were located. Therefore, the temperatures measured in this plane were fairly representative of slab temperatures that existed in any plane perpendicular to a wall at a point where baseboard radiation was located.

With one exception, the thermocouples in the kitchen floor slab were located in a plane perpendicular to the north wall and at a distance of 4 ft-10 in. from the baseboard radiation on the north wall of the living room and 4 ft-9 in. from the nearest radiation in the kitchen. Since the closest radiant baseboard was 4 ft-9 in. away from the plane of the thermocouple location, the temperatures measured by the thermocouples, shown in Fig. 4, were probably representative of slab temperatures that would have been obtained had no baseboard radiation been used.

A group of eight thermocouples was provided for the measurement of air temperatures at various levels in any room of the house. Four of these thermocouples were attached to a portable standard at levels of 3 in., 30 in., and 60

in. above the floor, and 3 in. below the ceiling. The other four thermocouples were similarly attached to a second standard. The lead wires to the thermocouples were of sufficient length to permit locating the standards in any room. The use of portable standards gave maximum flexibility in measuring air temperatures, and, when not in use, they could easily be stored in the utility room. A third group of thermocouples was provided to study performance of the component parts of the heating system, including the temperature of the water entering and leaving each circuit, and the temperature of the water entering and leaving the boiler. A chromelalumel thermocouple made of No. 22 B & S gage wire was located at the smoke outlet of the boiler.

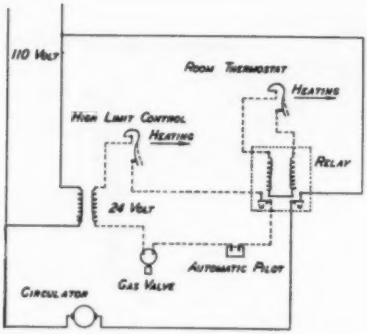

FIG. 5. CONTROL SYSTEM

Thermostat Calls for Heat: (1) Circulator and gas burner start; (2) Circulator runs until thermostat is satisfied; (3) Gas burner runs until thermostat is satisfied or until boiler water temperature reaches setting of high limit control. *Themostat Satisfied:* (1) Circulator and gas burner stop.

A central switchboard was located in the utility room and all of the thermocouple leads were connected to jacks on this board. Each hot junction was provided with an individual cold junction which was maintained at 32 F by an ice bath. All cold junctions were insulated electrically by wrapping with thread and coating with ambroid. A portable precision potentiometer and a 10-point recording potentiometer provided means for either instantaneous readings of the emf of each thermocouple or continuous printed records of the emf given by any copper-constantan thermocouple. The range of the 10-point recording potentiometer, however, was from -1 mv to $+5$ mv which did not permit recording of the flue gas temperatures.

A recording hygrometer was used to make continuous records of the air temperature and relative humidity at the 30-in. level in the living room. This instrument was checked periodically with an aspiration type psychrometer. The CO_2 content of the flue gases was checked periodically with an Orsat apparatus. Self-starting electric clocks were connected in parallel with the gas valve and

the circulator motor, and recorded the total time of the operation of each unit during the test period. A gas meter was used to measure the quantity of gas used.

TEST PROCEDURE

A gas burning rate of approximately 100 cfh was maintained during burner operating periods throughout the entire heating season. Burner adjustment was such that a CO_2 content of from 7 to 8.5 percent was obtained at the smoke outlet of the boiler. The natural gas used was supplied from the Texas-Oklahoma Pipe Line, and had a gross heating value of 1000 Btu per cu ft.

Since the house was occupied by a family consisting of a man, his wife, and one small child, cooking, washing and other domestic processes were carried on in a normal manner. Little attempt was made to control door positions between rooms. Although the occupants of the house generally kept the door between the kitchen and utility room closed, room air temperatures were not taken until the investigator assured himself that this door had been closed a minimum of 30 min prior to taking observations. Windows remained closed at all times. Observations of room air temperatures at 3 in., 30 in., and 60 in. above the floor, and 3 in. below the ceiling, were made at periodic intervals. Approximately every 48 hours, the operating time of the burner and the circulator, the cubic feet of gas consumed by the burner, and the floor slab temperatures were recorded. Periodic checks were made of flue gas temperature and CO_2. A continuous recording was made of the outdoor air temperature, the temperature of the water at the supply and return connections of the boiler and at the supply and return of each circuit. The relative humidity and air temperature at the 30-in. level in the living room were also recorded continuously.

During the main series of tests conducted during the 1947-48 heating season, the radiation in the bath was connected in series with the radiation in the two bedrooms to form one circuit of a one-pipe, forced-circulation hot-water system. The radiation in the living room was connected in series with that in the kitchen to form a second circuit. A few tests were run, during which the radiation in the bath was connected to a separate circuit, thus permitting the turning off of radiation in the bedrooms at night. This series was abandoned, however, when tests in the I=B=R Research Home indicated that no fuel savings would result from this practice.

In all tests, the thermostat was set to maintain an air temperature of 72 F at the 30-in. level in the living room both day and night.

RESULTS

Slab Temperatures: Fig. 6 shows the variation in surface temperatures of the floor slab over a range of indoor-outdoor temperature differences. The upper curve shows that essentially no difference existed between the average surface temperatures at the approximate centers of the living room and kitchen. Further examination of this curve shows that average surface temperatures at the center of the kitchen and the living room remained practically constant at 70.5 F over the range of the indoor-outdoor temperature differences from 0 to 40 deg. For indoor-outdoor differences greater than 40 deg, the average surface temperature decreased approximately 1.0 deg for each 10 deg increase in indoor-outdoor temperature difference. Thus, at the design condition of 80 deg indoor-outdoor

temperature difference, the average surface temperature in the center of the room would be about 66.5 F.

The lower curves in Fig. 6 show that slab surface temperatures, one foot from the west wall in the living room and one foot from the north wall in the kitchen, were markedly different. As a result of higher water temperatures in the heating system in the colder weather, the floor surface temperature in the living room increased slightly as the indoor-outdoor temperature difference increased. The average surface temperatures in the living room were 70.5 F and 73.0 F for indoor-outdoor temperature differences of 10 deg and 70 deg, respectively. In the kitchen, floor temperatures measured one foot from the north wall were influenced primarily by outdoor and indoor air temperatures. Hence, as the

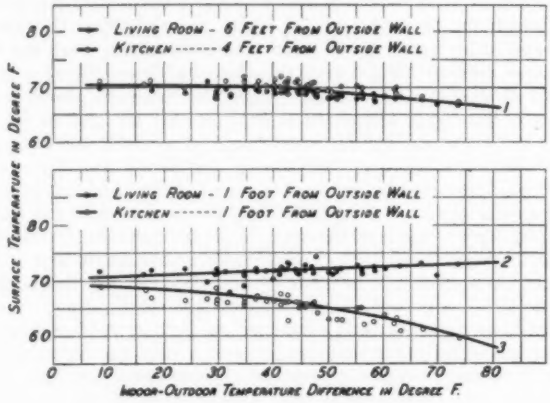

FIG. 6. SURFACE TEMPERATURES OF FLOOR SLABS

indoor-outdoor temperature differences increased, the surface temperature decreased. Average surface temperatures of 68.8 F and 61.0 F were obtained one foot from the wall in the kitchen at indoor-outdoor temperature differences of 10 deg and 70 deg, respectively.

An analysis of the data on the study of floor slab temperatures shows that the radiant baseboard is particularly adapted to maintaining comfortable floor temperatures in a basementless type of structure. Surface temperatures in the center of the room were always comfortable, and, wherever radiant baseboard was installed, temperatures near the edge of the slab were slightly higher than in the center of the room. Where no radiant baseboard was installed, however, floor temperatures near outside walls were considerably lower. Hence, the advantage of radiant baseboard lies in the large amount of outside wall exposure that a long, low unit of this type covers.

Room Air Temperature: In Fig. 7, the differences between the average temperatures of the air in the center of the living room, as measured at the 30-in. level, and at 3 in. above the floor, and 3 in. below the ceiling, are shown plotted against the indoor-outdoor temperature differences. The scattered pattern of

the points is attributed in part to the limitations in obtaining these temperatures with the house occupied. To prevent inconvenience to the occupants, the portable standard was set up each time a set of temperature measurements was to be taken. The readings were generally recorded for a duration of 30 min and without regard to burner and circulator operation.

Fig. 7 shows that, in the basementless house, the temperature difference from floor to ceiling increased with indoor-outdoor temperature difference. At 69.5 indoor-outdoor temperature difference, the coldest weather at which these temperatures were measured, a difference of only 4.6 deg was observed between the average temperature of the air 3 in. above the floor and 3 in. below the ceiling. For all practical purposes, this was about the same as that obtained with type RC radiant baseboard in the I=B=R Research Home[3]. The Research Home had a full basement.

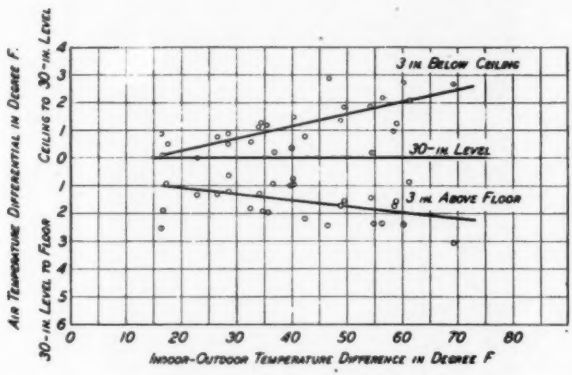

FIG. 7. AIR TEMPERATURE DIFFERENTIALS IN LIVING ROOM

Average air temperatures in the center of the living room of the basementless home, as measured 3 in. above the floor, were approximately 70 to 71 F for all indoor-outdoor temperature differences encountered. In the occupancy zone, 30-in. level to floor, temperature differences were only 1.0 to 2.0 deg. These results, together with results previously obtained in the I=B=R Research Home[3], indicate that the radiant baseboard, if properly installed, will maintain excellent air temperatures in homes built with basements, or in those built on concrete slabs in direct contact with the ground.

Fig. 8 is a reproduction of a seven-day recorder chart taken from an instrument located adjacent to the thermostat in the living room. During the course of the seven days that this chart was being made, three different settings of the thermostat differential were used. The chart illustrates clearly that the maximum and minimum air temperatures were affected by the length of the *on* and *off* periods of the burner and circulator. Long cycles of burner and circu-

[3] A Study of Radiant Baseboard Heating in the I=B=R Research Home. (University of Illinois, *Engineering Experiment Station, Bulletin* 358, p. 22.)

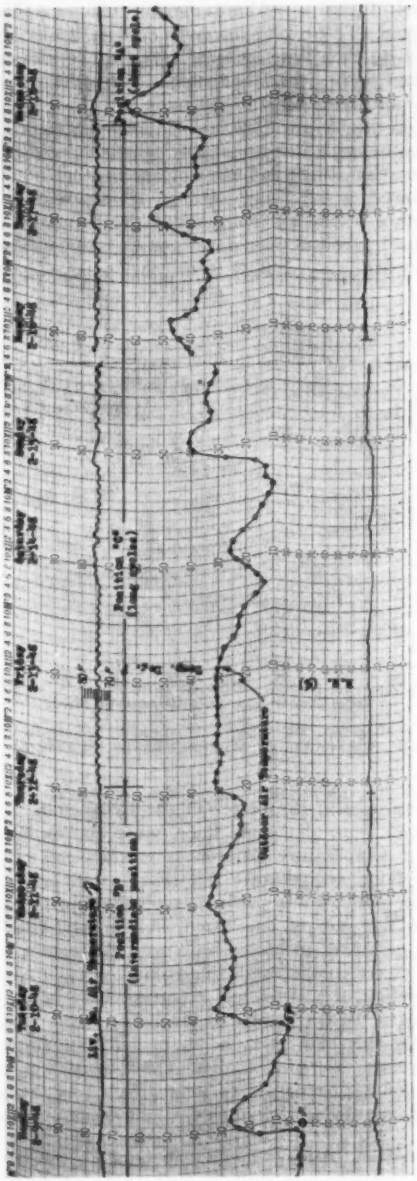

FIG. 8. AIR TEMPERATURE AT THE 30-IN. LEVEL IN THE LIVING ROOM

lator operation, obtained by setting the adjustable differential on the thermostat to position *C*, resulted in cyclic air temperature variations between *on* and *off* period of approximately 1.5 deg. Intermediate cycle length, obtained with the position of *B* setting, gave a variation of about 1 deg, and short cycles, obtained with the position of *A* setting, gave practically no variation in the living room air temperature at the 30-in. level. Air temperature variations near the ceiling were slightly greater than those just referred to. Furthermore, the temperature variations in the living room were more pronounced than in the other rooms. Without the aid of thermometers, the occupants of the house could not detect cyclic air temperature variations of 1.5 deg.

Fig. 8 also shows that, aside from the variations in air temperature discussed in the preceding paragraph, the outdoor temperature had no effect on the indoor air temperature at the thermostat level. The indoor air temperature remained

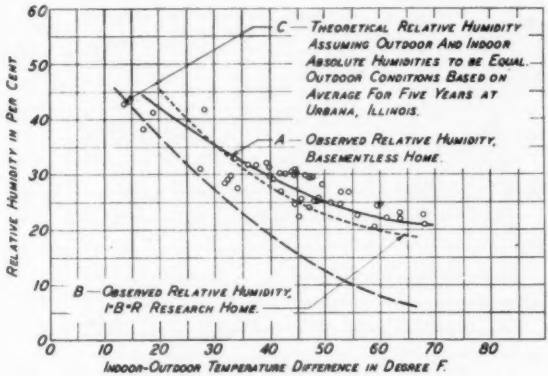
C — Theoretical Relative Humidity Assuming Outdoor And Indoor Absolute Humidities to be Equal. Outdoor Conditions Based on Average For Five Years at Urbana, Illinois.
A — Observed Relative Humidity, Basementless Home.
B — Observed Relative Humidity, I-B-R Research Home.
Relative Humidity in Per Cent
Indoor-Outdoor Temperature Difference in Degree F.
0
10
20
30
40
50
60
70
80

FIG. 9. INDOOR RELATIVE HUMIDITY

constant, even though there was a large and sometimes rapid change in the outdoor temperature. Thus, it is demonstrated that a hot-water system using radiant baseboard can take care of sudden changes in load with little difficulty.

The air temperatures at the 30-in. level in the living room were approximately 1.5 to 2.0 deg higher than those in all other rooms, except the bath. The radiation requirement for the bath room, however, was based on an 85 deg indoor-outdoor temperature difference, instead of the 80 deg temperature difference used for the other rooms. Air temperatures in the middle of any room, as measured at the 60-in. level, were approximately 0.5 deg higher than the temperatures measured at the 30-in. level.

Humidity: Humidity observations were made in the basementless home during the 1947-48 heating season, even though no attempt was made to increase the indoor relative humidity with a humidifier. Curve *A* in Fig. 9 shows the relative humidity observed at the 30-in. level in the living room of the basementless home; curve *C*, the relative humidity which would have existed in the house if the absolute humidity indoors had been the same as that outdoors;

and curve *B*, the relative humidity observed in the I=B=R Research Home[4] with no special means of providing humidification. Each test point represents the average relative humidity for a 48-hour period. The difference between curves *A* and *C* can be considered as representing the moisture added to the indoor air by cooking, by washing, by the occupants, by evaporation from building materials and house furnishings, and by evaporation from water normally used for general purposes in the house, minus that lost to the outside air.

It may be noted that the relative humidities observed in the basementless home and the I=B=R Research Home, as shown by curves *A* and *B*, are in fairly close agreement, yet no cooking or washing was done in the Research Home.

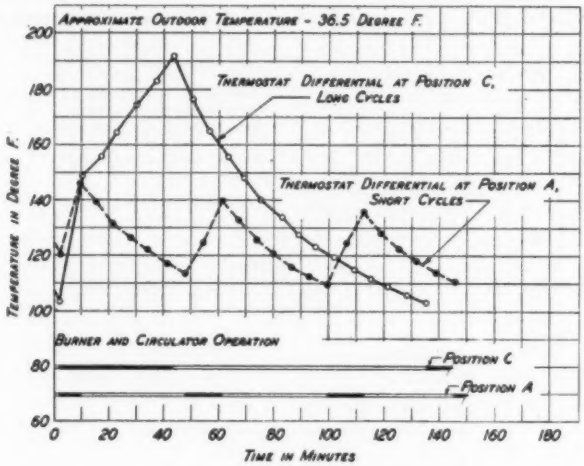

FIG. 10. BOILER INLET WATER TEMPERATURES, TWO POSITIONS OF THERMOSTAT DIFFERENTIAL ADJUSTMENT

Although both houses were insulated and provided with vapor barrier paper in the exposed walls and ceiling, and weatherstripping was applied to the doors and windows, the basementless home was not as tight in construction as the I=B=R Research Home. In the basementless home, the outside doors and the butted joints of the prefabricated wall panels were not tight, and possibilities existed for considerable air leakage and vapor transmission. Thus, it is reasonable to assume that, if the basementless home was of tighter construction, higher relative humidities would have been obtained. The problem of maintaining adequate indoor humidities for comfort in winter cannot be separated from consideration of good building construction. If houses are tightly constructed, little difficulty is experienced in maintaining an adequate indoor humidity in winter.

[4] Performance of a Hot-Water Heating System in the I=B=R Research Home at the University of Illinois. (University of Illinois, *Engineering Experiment Station, Bulletin* 349, Fig. 27.)

Water Temperatures in Boiler and Radiant Baseboard: As shown in Fig. 10, position *A* of the adjustable differential on the thermostat gave shorter *off* and *on* periods, and, hence, more cycles of burner and circulator operation for a given indoor-outdoor temperature difference than did position *C*. It may also be noted that boiler inlet water temperatures for position *A*, at 36.5 F outdoor temperature, varied from 110 F to about 140 F, a range of 30 deg, while water temperatures for position *C* varied from 102 F to 191 F, or a range of 89 deg. These ranges of water temperatures at the boiler inlet are fairly indicative of changes in water temperatures in the baseboard radiation. Thus, with the shorter cycles of burner and circulator operation, smaller fluctuations occurred

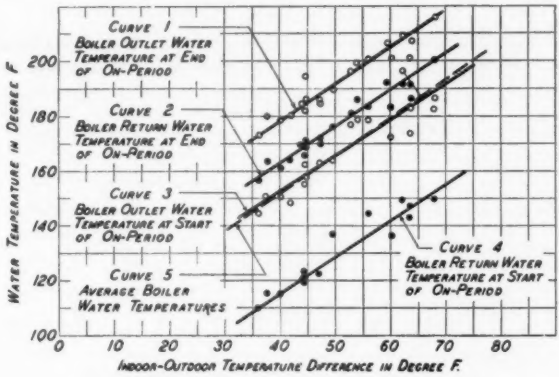

FIG. 11. BOILER WATER TEMPERATURES

in the water temperatures for the radiant baseboard and, hence, smaller variations occurred in the indoor air temperatures.

In Fig. 11, curves *1* and *2* show the water temperatures obtained at the boiler outlet and return connections at the end of the circulator *on* period, whereas curves *3* and *4* show the water temperatures at the outlet and return connections at the beginning of the circulator *on* period. The arithmetical average of curves *1, 2, 3,* and *4* is represented by curve *5*. For all practical purposes curve *5* represents the average temperature of the water in the boiler for any given indoor-outdoor temperature difference. It is also representative of the average water temperature in the radiant baseboard.

The radiation installed in the basementless home was just sufficient to offset the calculated heat loss at an outdoor temperature of —10 F when the radiant baseboards were supplied with an average water temperature of 215 F. By extrapolation of the boiler water temperature curve *5* in Fig. 11, it will be found that, at an indoor-outdoor temperature difference of 80 deg, the average boiler water temperature was 205.5 F, and the average water temperature in the radiant baseboards, which was essentially the same as the average boiler water temperature, would have been about 9.5 deg lower than that selected for the design of the system. Thus, the amount of radiation installed was sufficient to carry the load at design conditions with only a small reserve capacity available.

Table 3 shows the temperature drop through the boiler and radiator circuits with both a two-circuit and a three-circuit flow. In the two-circuit system, the bath was included in the bedroom circuit. In the three-circuit system, the bathroom radiation was in a circuit by itself. It may be noted that the maximum temperature drop that occurred in any circuit was 13.5 F. Since the heating system was designed on a 20 deg temperature drop, it is evident that the flow of water in each circuit was more than ample to take care of the radiation connected to the circuit.

Fuel Consumption: Fig. 12 shows daily burner and circulator operating time and fuel consumption for operation at three settings of the thermostat differential. While changes in the thermostat differential had a marked effect on

TABLE 3. TEMPERATURE DROP THROUGH BOILER AND RADIATOR CIRCUITS WITH SYSTEM OPERATING NEAR DESIGN WATER TEMPERATURE

CIRCUIT	AVG INLET TEMP F	AVG OUTLET TEMP F	TEMP DROP F DEG
TWO CIRCUITS			
Boiler	197.8	210.0	−12.2
Bath-Bedroom	209.5	197.7	+11.8
Liv Rm.-Kitchen	209.3	199.5	+ 9.8
THREE CIRCUITS			
Boiler	196.5	206.5	−10.0
Bath	205.9	202.7	+ 3.2
Liv. Rm.-Kitchen	206.0	194.5	+11.5
Bedroom	206.0	192.5	+13.5

the length and number of operating cycles for any given indoor-outdoor temperature difference (Fig. 10), it may be observed in Fig. 12 that the setting of the thermostat differential adjustment had no appreciable effect on the total daily operating time of the burner or circulator or on the daily fuel consumption.

By extrapolation of the curve for burner operating time, shown in Fig. 12, a total operating time of only 13.5 hours per day would be obtained at an indoor-outdoor temperature difference of 80 deg, thus indicating that the burner and boiler had ample reserve capacity.

Since the calculated heat losses of the Research Home and the basementless house were in close agreement, the fuel consumption obtained in the Research Home, when using radiant baseboard, has been included in Fig. 12, and, as might be expected, the daily fuel consumption for the two houses was also in close agreement at all indoor-outdoor temperature differences.

Cleanliness of Operation: The basementless home was ready for occupancy in May 1947. When heat was applied, all the walls were freshly painted and all the curtains and draperies were clean and new. After operating through a complete heating season of approximately nine months, faint dirt patterns were observed on some of the walls above the radiant baseboard. Curtains and

draperies that hung over the radiant baseboard, however, were as clean as similar curtains and draperies under which no radiant baseboard was located.

Dirt patterns are the result of dust being carried close to the walls by convection currents and being deposited by thermal precipitation. Experience in the

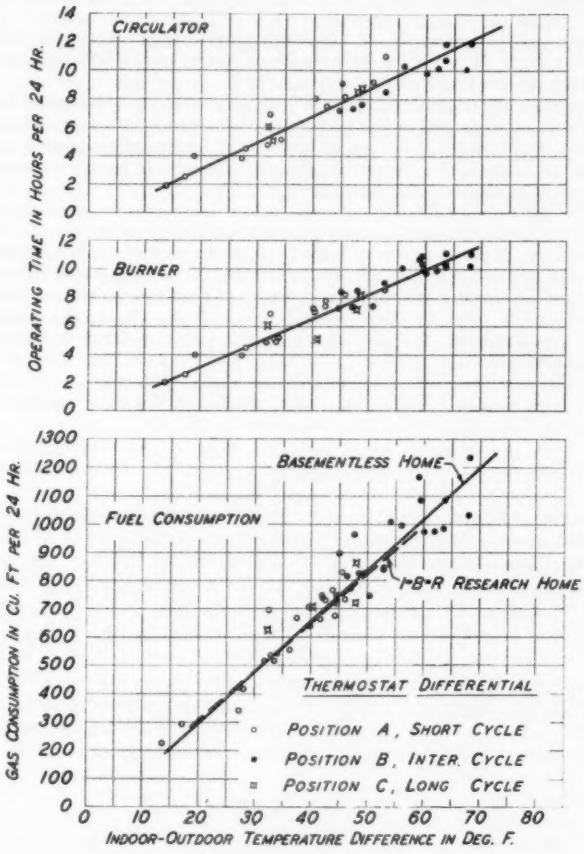

FIG. 12. BURNER AND CIRCULATOR PERFORMANCE

I=B=R Research Home and elsewhere has indicated that dirt patterns may be eliminated by reducing the design water temperature in the radiant baseboard to 200 F or lower. This decreases the convection currents set up by the radiant baseboard, and, hence, reduces the magnitude of the temperature gradient near the wall.

SUMMARY AND CONCLUSIONS

A summary of the test results of this investigation and the conclusions that may be drawn are as follows:

1. The radiant baseboard is particularly adapted to maintaining comfortable floor slab temperatures in a basementless structure, since long, low units of this type cover a large percentage of outside exposure.
2. Average air temperatures, as measured three inches above the floor, were approximately 70 to 71 F for all indoor-outdoor temperature differences encountered when the temperature at the 30-in. level was 72 F.
3. Temperature differences between the occupancy zone, as measured 3 in. and 30 in. above the floor, were only of the magnitude of a degree and a half to two degrees.
4. The hot-water heating system using radiant baseboard responded quickly to sudden changes in load, thus maintaining constant room air temperature even while the outdoor temperature was changing rapidly.
5. The problem of maintaining adequate indoor humidities for comfort in winter cannot be separated from consideration of good building construction. At 10 F outdoor temperature, 22 percent relative humidity was obtained in the basementless home.
6. Fuel consumption was not affected by setting of the adjustable differential on the thermostat. However, water temperatures in the radiant baseboards, and consequent fluctuations in room air temperatures, were affected. The longer lengths of the *on* and *off* periods of the burner and circulator resulted in greater fluctuations of room air temperatures.
7. Faint dirt patterns were observed on some of the walls above the radiant baseboard after nine months of operation. Dirt patterns of this type can be eliminated by limiting the water temperature to a maximum of 200 F.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the investigation of steam and hot-water heating systems being made in the Department of Mechanical Engineering at the University of Illinois. The investigation is a cooperative project sponsored jointly by the Engineering Experiment Station of the University and the *Institute of Boiler and Radiator Manufacturers.*

This paper is largely an abstract of a thesis written by Mr. Weigel in partial fulfillment of the requirements for the Degree of Master of Science in Mechanical Engineering in the Graduate College of the University of Illinois. Also, these results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is hereby made to the various manufacturers who cooperated by furnishing materials and equipment.

DISCUSSION

G. S. MACLEOD, Chicago, Ill.: I would like to ask: (1) Were the radiant baseboards in the test house covered by cabinets or fixtures during any of the tests? (2) What effect, if any, would this have on the emission of radiators so placed? (3) Has any determination been made of the relative amounts of heat transfer by radiation and by convection of the type *RC* and type *R* radiators?

E. R. TESKE, Chicago, Ill.: I think the authors are to be complimented for a fine paper. However, I feel that the last paragraph of their conclusion is open to further consideration and possibly a few questions. They state that smudging can be eliminated with RC type or radiant baseboard by limiting the water temperature to 200 F or less. Actually, this is not apparently backed by test data.

We have a local contractor who has done a number of installations in that field and was particularly concerned about smudging, since he had installed over 1000 units and some were creating complaint calls. As no information was available, he made a test installation in his own shop. An exposed 12-in. brick wall, furred and plastered, served as the *test rack* for the baseboard units being compared. Insulation was provided in the furring space. The floor was a concrete slab on the ground. The room had a large metal casement window above the units under test. It was a pipefitting storage area and the concrete floor was bare. Consequently, dirt and dust conditions were severe.

This installation was operated continuously for a period of 90 days at a minimum of 215 F water temperature. The test rack had three convector-type baseboard units installed in series. He found that the determining factor in smudging, as indicated by visual comparison and recorded by photographs, was definitely the physical shape of the enclosure, notably the position of the air outlets. The front outlet types were definitely superior to the top or slanted outlet designs in the elimination of smudging characteristics.

So it seems that, while the water temperature has some influence, it is minor in comparison to the actual physical characteristics of the air pattern from the baseboard units. In view of the large use that these units are finding in competitive types of construction, it is an economic burden when you ask that the system be designed for less than 215 F water.

There is also the point in mind that a survey was made of Chicago weather conditions for the last twenty years, and, during the normal heating season, it was found that there were only 3 percent of the days which required over 200 F water if a modulating hot-water system designed on 240 Btu output was used.

Mr. Ljunggren, the contractor who made this investigation, feels that water temperature is not the answer and that, if such a statement were to find its place in THE GUIDE, eventually it would condemn as unsound engineering practice all installations that had been made on 240 Btu output, and also would present economic limitations to the application of this type of heating to the low-cost housing now in demand and in which baseboard could be installed quite cheaply.

I would like the authors to enlarge on the paragraph which more or less relates to the cleanliness of the system. It says *other field installation,* or something to that effect. Maybe the authors can explain further why they gave 200 F water temperature limitation as the answer.

AUTHORS' CLOSURE: We will attempt to answer Mr. McLeod's questions first. In order to place all of the radiant baseboard in the kitchen along an outside wall, it was necessary to install a 3-ft section directly behind the stove. Any effect on output due to the obstruction was not apparent as the room heated satisfactorily. The radiant baseboards in the bath and rear bedroom, which was actually used as a nursery, were entirely free of obstructions. In the front bedroom and living room, however, furniture and draperies or curtains partially covered the heating units. Furniture placement was entirely at the discretion of the occupants. The draperies in the living room extended to the floor, and, when pulled, entirely covered the section of radiant baseboard directly beneath the windows. Although there were times when the draperies remained pulled, there was no noticeable effect on the comfort of the occupants.

As far as the relative values of heat transfer by convection and radiation from baseboard heating units are concerned, no work has been done at the University of Illinois to determine these various percentages.

Mr. Teske requested further discussion on the last conclusion of our paper which deals with dirt patterns. The design temperature of the water in the radiant baseboard in the basementless home was 215 F. This was on a minus 10 outdoor temperature and a 70 F indoor temperature. Since some minus 5 F temperatures occurred in

Urbana during the 1947-1948 heating season, the heating system did operate near design conditions. These high water temperatures and resultant convection currents no doubt contributed to the formation of dirt patterns.

Photographs of the baseboard and the wall directly above in the basementless house were made before the heating system was turned on and again after nine months of operation. In the kitchen, where one is likely to have more dirt and a greater disturbance of that dirt or dust, faint dirt patterns were formed. The character of these patterns were of this nature. Directly above the long slot in the front of the RC unit, the dirt deposits on the wall are fan-shaped. On the section of wall between the two slots, the smudging appears as an inverted fan.

Types R and RC radiant baseboards were also tested in the I=B=R Research Home. These systems were also designed on a water temperature of 215 F. Faint dirt patterns with the type RC baseboard were observed, but the pattern was not as pronounced as that obtained with small-tube radiation.

We can appreciate Mr. Teske's concern over placing a limitation on the design water temperature, but it is true that, with any given design of radiator, the lower the operating temperature, the lower the tendency to cause dirt streaks on walls.

We do agree with Mr. Teske that the design of the unit is an important factor in the cause of dirt patterns. Our tests were all made with a front-outlet type RC unit which admittedly is superior to the top-outlet design as far as cleanliness of operation is concerned. It is possible that some change in design of the units tested would help to reduce further the formation of a dirt pattern.

1359

No. 1359

HEATING PANEL TIME RESPONSE STUDY

By A. B. ALGREN* AND BEN CISCEL**, MINNEAPOLIS, MINN.

THE problem of securing optimum comfort through automatic controls is often considered only after a house and its heating system are designed and built. Ordinarily such a heating system is designed or chosen on the basis of cost and its ability to supply the *steady-state* heat losses for the structure. The control problem is then a matter of selecting control equipment which is suitable to the rest of the installation, but which must also provide maximum comfort per dollar. The increasing number of homes being constructed with floor panel heating systems led to this study of the heat storage effects in the controlled system. In this study, heat input rates and losses were an important consideration.

EXPERIMENTAL HOUSE

These tests were conducted in a small, well-insulated house (Fig. 1) without windows, and with only one door. The heating panel was a 4-in. concrete slab laid on 9 in. of crushed rock, with periphery insulation of rigid fiber board. The general arrangement and piping layout are shown in Fig. 2.

During the tests, only natural ventilation was used. The panel circulating water was heated by a heat exchanger connected to a steam system. The complete control system, as shown in Fig. 3, permitted considerable freedom in the choice of tests to be performed. The controller had two main sections: (1) for the hot water temperature, which depended upon the steam input to the heat exchanger; and (2) for determining the amount of hot water to be mixed with the recirculating water as a function of inside and outside temperatures.

More than a hundred points of temperature measurement, including the floor panel, were variously located inside the house. The room air temperature was measured by a system of wire thermocouples mounted about 4 in. from each wall and shielded from direct panel radiation. The average of these readings was taken as *room air temperature*. The *mean radiant temperature* was determined by networks of 12 thermocouples taped to each of the wall, ceiling and floor surfaces. The reading for each surface was weighted according to the area involved in calculating the mean radiant temperature.

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** Control System Analyst, Minneapolis-Honeywell Research Division, Minneapolis-Honeywell Regulator Co.

Presented at the 55th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1949.

Studies of random temperature variation in this and other houses led to the conclusion that ordinary control action produced temperature changes too small to be accurately traced in terms of time response. Hence, the investigators decided to produce large temperature changes artificially, so that all the quantities involved would be measurable. Even though heat losses are not linear with mean temperature, it was considered better to have large readings with possible errors up to 5 percent, rather than to have small readings difficult to interpret.

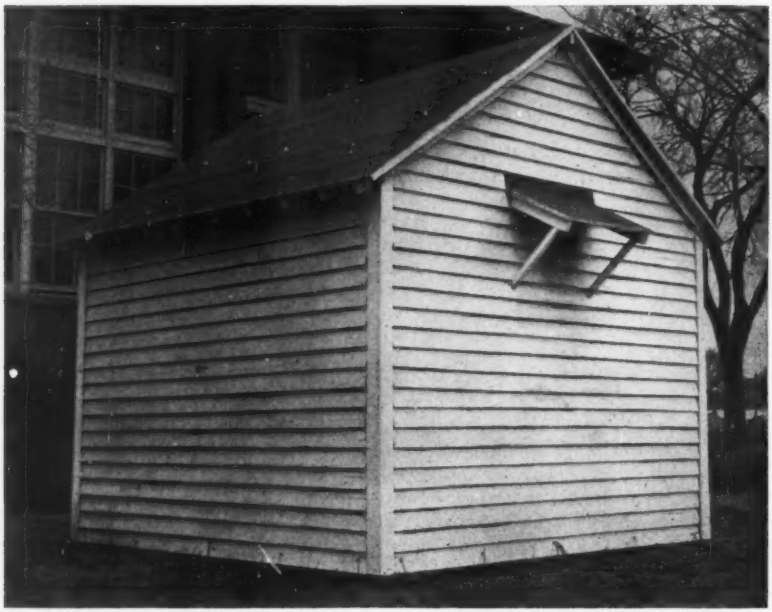


FIG. 1. EXPERIMENTAL HOUSE, UNIVERSITY OF MINNESOTA

The transient conditions were initiated as follows. With the house under ordinary *steady-state* control, the inside controller was turned off and the recirculating valve was set for maximum hot water input to the panel. During the test, one or more recorders were used to record temperatures at a number of points of interest (up to 48 points, in some cases).

TRANSIENT RESPONSES

The primary factor in house temperature control is the time lag of the heating panel. The authors were interested, however, in the transient conditions occurring in the *complete heating system*. Thus, the investigators began with the artificial demand for heat, and then measured water, panel, wall, and air

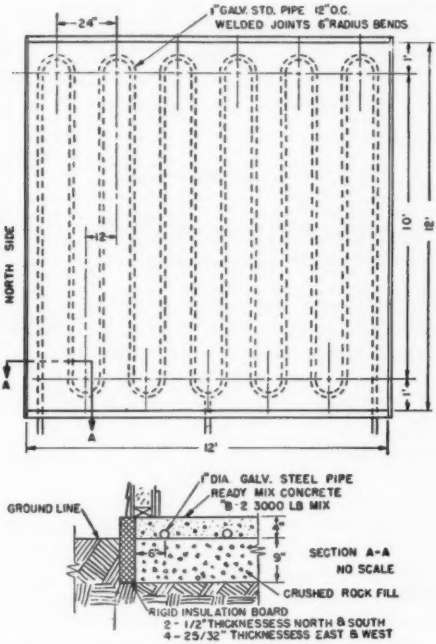


FIG. 2. FLOOR PANEL CONSTRUCTION DETAILS

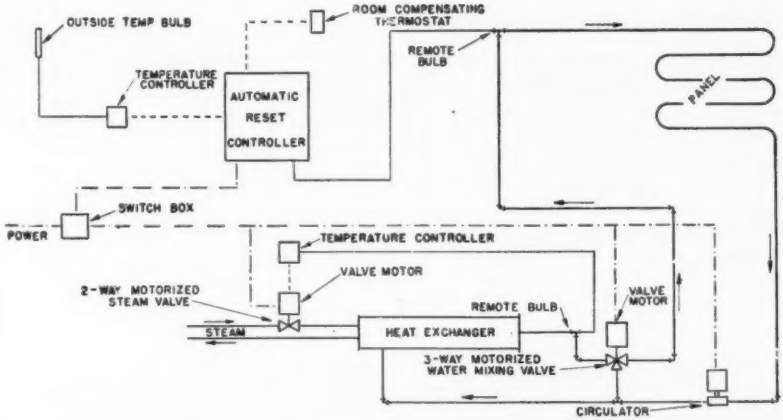


FIG. 3. HEATING SYSTEM WITH CONTROLS

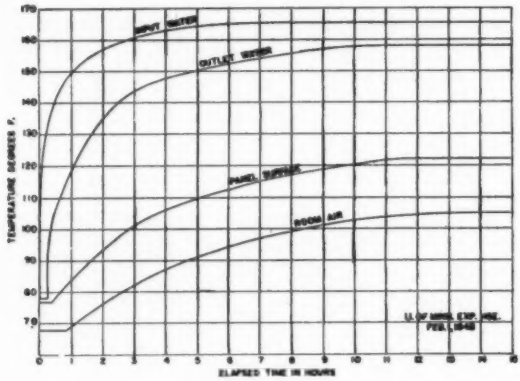


FIG. 4. TRANSIENT RESPONSES WITHIN THE HOUSE

temperatures. Some of these are shown in Fig. 4, with elapsed time as the independent variable.

In a study of the control problem, the actual values of temperature are not as important as the time required to reach the final (or some percentage of final) temperature. For that reason, the curves of Fig. 5 show the transient temperatures in terms of percentage change.

In the study of transient conditions, the times-of-response are often compared at the 63 percent mark; that is, where the temperature (or other variable) has gone through 63 percent of the difference between the initial and final values. Although these panel and house responses are more complicated than those produced by a single mathematical time constant, the responses can be compared at the 63 percent level. It will be noted in Fig. 5 that almost 4 hours were required by the panel to reach the 63 percent level.

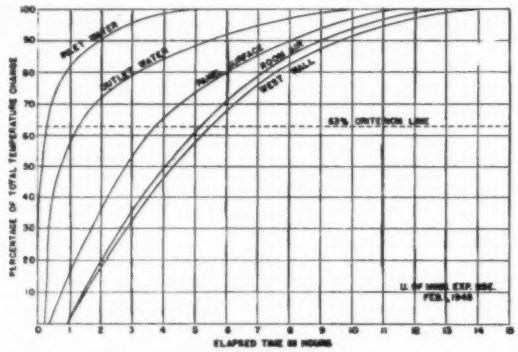


FIG. 5. TRANSIENT RESPONSES ON A PERCENTAGE BASIS

The unique aspect of the data of Fig. 5 is not the panel lag, but in the hour and quarter lag of the room air behind the panel surface at the 63 percent criterion line. The west wall is included in Fig. 5 merely to indicate the trend of surface temperatures during the transient conditions studied. The surface temperatures of the other walls and of the ceiling were about the same as those for the west wall.

TEMPERATURE MEASUREMENTS

The literature concerning comfort in terms of air and mean radiant temperature is extensive. For that reason, the experimental house was instrumented and constructed to facilitate measurements of surface temperatures and air temperatures.

The fact that the floor surface temperature was so far above room air temperature would lead most people to expect that the mean radiant temperature would be greater than the room air temperature in both time and magnitude. The curves of Fig. 6 show the actual temperatures for Feb. 1, 1948, in the well-

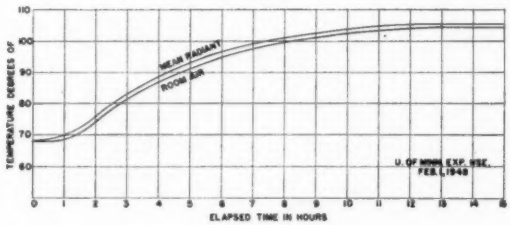


FIG. 6. MEAN RADIANT AND ROOM AIR TEMPERATURES

insulated test house, where the mean radiant and room air temperatures differed by about one degree.

Tests on other days tended to verify the fact that mean radiant temperature (MRT) and air temperature T_a were generally within a degree of each other, even during transient conditions as sudden and as large as those obtained in these tests. This MRT did not take into account the temperature of doors and window areas. In a well insulated home having normal percentages of window and door area, however, the results would not be significantly affected by the temperature of these surfaces. Another point to be considered is the lack of ventilation, except that due to infiltration. This is being studied further for inclusion in a future report. In addition, this investigation is being expanded to include the effect of large window areas.

HEAT INPUT RATES

In Fig. 5, the curves of inlet and outlet water temperatures vary widely from design values. Since water was constantly recirculated in the panel at a rate of 1.5 gallons per minute, the difference in temperature between inlet and outlet water was directly proportional to heat flow in Btuh.

Although the ordinary design calculations for a house of this type would indicate a design load of about 3000 to 5000 Btuh, the authors' transient heat inputs ran several times that amount. These flow rates are shown in Fig. 7, with the heat supplied to the panel appearing to be about six times that required to offset normal house losses. These large heat inputs could not be correlated

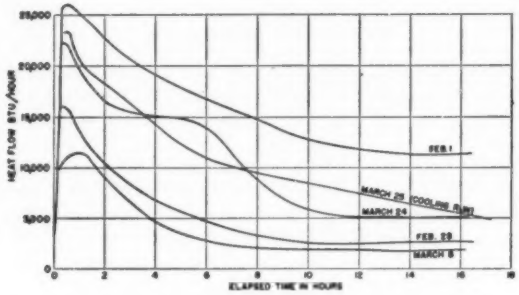


FIG. 7. HEAT FLOW RATES DURING TRANSIENT CONDITIONS

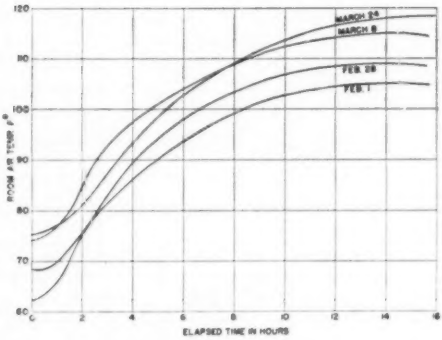


FIG. 8. ROOM AIR TEMPERATURES DURING TRANSIENT CONDITIONS

directly with inlet water, outside air, or floor surface temperatures. However, the room air temperatures (see Fig. 8) rose in succeeding tests. This seems to indicate a change in the storage or heat loss factor for the house after several heating cycles.

The trend of heat input rate (becoming progressively smaller) was reversed in the March 24 run. This reversal may have been due to a general thawing around the perimeter of the panel. (A study of temperatures on the preceding days showed a minimum temperature of —26 on March 11, although the average outside temperature for the week preceding the date of the test was +39 deg.)

A cooling run is also shown (March 25, Fig. 7) giving an indication of heat flow back out of the panel after a heating run on the preceding day. This was accomplished by running cool water through the panels. The cooling run (Fig.

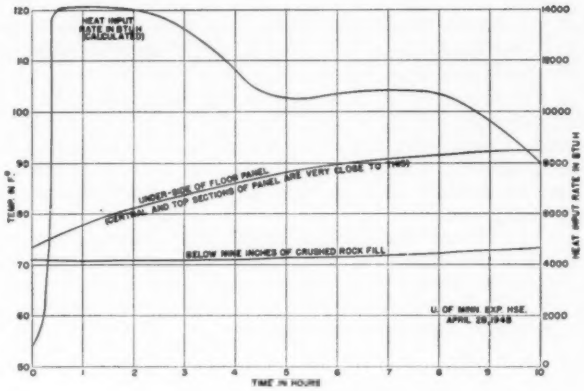

HEAT INPUT RATE IN BTU/H (CALCULATED)
UNDER-SIDE OF FLOOR PANEL (CENTRAL AND TOP SECTIONS OF PANEL ARE VERY CLOSE TO THIS)
BELOW NINE INCHES OF CRUSHED ROCK FILL
U. OF MINN. EXP. HSE. APRIL 25, 1948
TEMP IN °F
HEAT INPUT RATE IN BTU/H
TIME IN HOURS

FIG. 9. TYPICAL TRANSIENT TEST CONDITIONS

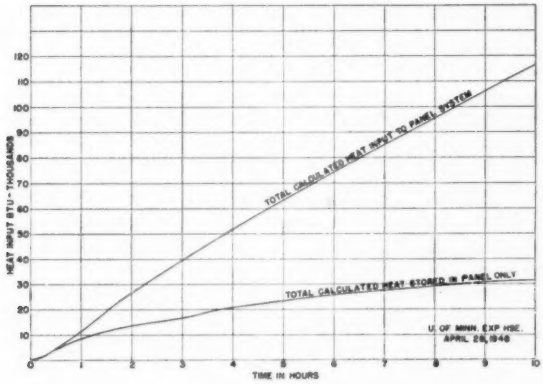

TOTAL CALCULATED HEAT INPUT TO PANEL SYSTEM
TOTAL CALCULATED HEAT STORED IN PANEL ONLY
U. OF MINN. EXP. HSE. APRIL 25, 1948
HEAT INPUT BTU - THOUSANDS
TIME IN HOURS

FIG. 10. TIME INTEGRAL OF HEAT INPUT RATES

7) indicates that much of the heat input to the panel was stored below ground and was available for return, depending upon the temperature level. However, much of this energy would probably have been lost in the earth if the cooling run had not been made.

The rate of heat input (shown in Fig. 7) brings up the question of total heat input to the panel system during a given period of time. For example, the data of Fig. 9 show a sudden heat input to the panel, with the calculated

heat input rate reaching a maximum of about 14,000 Btuh. Also shown in Fig. 9 are the average temperatures at the underside of the floor panel and at the bottom of nine inches of crushed rock fill. The temperature at the bottom of the crushed rock fill (under the panel) showed little change even after 10 hours.

Attempting to account for these heat inputs, the authors calculated and plotted the storage in the panel (Fig. 10). The large difference between total heat stored in the panel and total heat supplied to the panel system indicated that panel storage accounted for only a part of the total heat input. The house thermal load accounted for about 2500 Btuh or a total of 25,000 Btu over the 10-hour period. About 50,000 Btu then remained to be accounted for by storage in the fill or structure, and by periphery losses. The need for insulation between the panel and the rock fill, as a means of improving the temperature response of the panel, is apparent.

At the end of 10 hours, the total heat supplied to the panel system was about 116,000 Btu, of which 25,000 had been transferred into the house. This indicates that, if some system failure should cause a panel to be cooled, only a fraction of the heat supplied on resumption of heating would go into the house until the panel and fill came up to temperature.

CONCLUSIONS

1. The tests indicated that the mean radiant temperature was generally within a degree of the room air temperature.
2. The maximum air temperature rise rates were about 8 deg per hour and usually occurred about an hour after the heat input had been increased. Floor coverings or a smaller heat-input to the panel would further delay the rise of room air temperatures.
3. Heating plants for floor-panel heated houses should be capable of supplying heat well above the expected demand in order to take care of transient heat storage and losses.
4. Measurements taken before and after these periods of transient conditions show that *steady state* losses were within reasonable limits as contrasted with the excessive heat inputs required during heating up periods.
5. The air temperature was higher than the wall and ceiling temperatures. This indicated some heat transfer to these surfaces by convection and conduction as well as radiation.
6. The transient conditions in these tests were much larger than those which would be encountered in normal control operations, but the time lags involved should be independent of the temperature variation range.

ACKNOWLEDGMENTS

The authors wish to extend their gratitude to the University of Minnesota and Prof. F. B. Rowley, director of the Engineering Experiment Station, University of Minnesota, for cooperation in making available the experimental house for these tests. Robert Atkinson and Don Vandergon worked as University Research Assistants on the project. Many engineers of the Minneapolis-Honeywell Regulator Co. contributed ideas which were helpful, and especial thanks are due to W. H. Kliever, director of research, and R. T. Squier, asst. director of research, for their guidance and assistance.

DISCUSSION

E. A. JONES, Dayton, Ohio: The information presented in this paper is valuable, interesting, and timely.

I am particularly interested in paragraph 2 of the Conclusions, reporting a temperature rise of about 8 deg per hour. To get control, it is necessary to control the addition of heat in the house, but a reversal of the temperature trend outside may require rapid reduction in the heat supplied. I would like to inquire whether the authors found, because of the large amount of heat storage, that the drop was of the magnitude of 8 deg per hour or more, or less. My own experience is that it might be even slower than the rise.

Mr. Ciscel suggests that because of the large input required to speed response, the heating equipment in panel-heated jobs of this sort should be rather oversize in comparison to other systems. What consideration has been given to taking care of the reversal in temperature? Carrying his reasoning to the other extreme, it would appear that, to secure good control, it would perhaps, be advisable to leave off storm sash and insulation. Will you comment on this point?

The ratio between heat loss and thermal mass of heating equipment, including the floor slab or panels, is also important. Outdoor controls cannot predict sufficiently in advance rapid changes in heat demand when response is slow, if there is a rapid rise in outdoor temperature with sun effect through glass. A formula which could be used to avoid uncontrollable systems due to too great mass ratio would, I believe, be helpful to the industry.

A. S. CHIPLEY[1], Libertyville, Ill.: My question also is directed to Mr. Ciscel. We are conducting a series of tests on a radiant system whereby we suspend a series of horizontal pipes from the ceiling and attach aluminum facing panels to those pipes to form a suspended ceiling. Hot water is circulated through the pipes. The surface temperature of this ceiling responds quickly to changes in circulating water temperature. We are presently controlling the system (it is in our drafting room) by an ordinary room thermostat, which, currently, is cutting the circulating pump on for about 10 min and then off for roughly 20 min.

We know from measurements that the ceiling cools off rapidly after the water is shut off. We also know that the room feels comfortable—no variation in temperature is felt. Recording thermometer charts taken at head and floor levels show no variation in temperature.

My question is: What are we measuring? We know that the radiant emissivity of the ceiling is high for a part of the time, after which the ceiling rapidly cools to room temperature (or below, since we are also using the ceiling area to introduce fresh air). Yet our air temperature is uniform, according to recording charts. Does the uniform chart necessarily mean comfort?

G. S. MACLEOD, Chicago, Ill.: This question is also addressed to Mr. Ciscel. In your paper, you describe the method of using 12 thermocouples taped to each wall surface in finding the mean radiant temperature of the room. What correlation exists between this method and the use of such instruments as the eupatheoscope or the globe thermometer?

AUTHORS' CLOSURE (Mr. Ciscel). The first question was that of the correlation between the globe thermometers and the system of wall surface thermocouples that we use. I do not know of any. There may be in some instances.

When we started out, we had a complete system of thermocouples. It seemed the natural tendency, therefore, to continue with what we had, and so we put in a system of thermocouples on the walls. Perhaps, the method was approximate. Nevertheless,

[1] Chief Development Engineer, Burgess-Manning Co.

the data obtained from 12 thermocouples in a room of this size were considered satisfactory. No correlation with the eupatheoscope reading was attempted.

In regard to Mr. Jones' question, the data did not disclose the entire procedure. If you refer to Fig. 7 of the paper, you will see that, at one time, March 25, we ran a cooling run. We had been running these heating runs during which we ran in hot water. It seemed like it might be interesting to turn around and run cooling water in and to see how much heat we could recover from the panel system. As you can see, we did fairly well. The amount recovered depends upon the temperature of the water circulated. There is, nevertheless, a possibility of running in cooling water. Of course, if you have the same temperature differential between the water that you run in to heat and the water you run in to cool, you can get the same rate of change of room air temperature.

As to the suggestion of leaving insulation off and so on, I think it has merits in some respects. It seems to be the standing joke of people in the control business, to ask the occupants, *How is the control system working?* They say, *Oh, fine. It does not overshoot or anything.* And then they add, *If it gets too hot, we just open the windows.*

This is really an indication that the control system has broken down. Yet they think it is adequate because all they have to do to correct the condition is to open the windows. Obviously, we do not feel that this is an answer. If you install a control system, it is supposed to be a control system, whether the windows are open or closed.

The control system shown in Figs. 2 and 3 does control this system. We have a second one, developed this past winter, that is much simpler. It also does a good job and is one used in the tests lately. I can say definitely that we do not have to open windows to make a good control system work, even though the possibility of dropping the panel temperature is a difficult problem.

Mr. Chipley asked about what he was measuring. That is always a problem.

First of all, let me make one statement about the system under control and the control system. These are two different items. In this instance, the system under control is a panel system with a source of hot water. The controlled system includes the panel and the controls. In other words, the thermostat on the wall, the reset device, the valves and all the rest, and the responses obtained from the controlled system are not necessarily the same as those you get from the panel alone. In this paper, we investigated the panel alone. It is a separate investigation to see how well the control systems control. We are not disclosing that information as a part of this paper. All I can say now is that it does and we have curves to show you, if you are interested.

In answer to whether Mr. Chipley was measuring comfort within the room, I think that is something which extends beyond the realm of engineering. Maybe that is a strong statement. My own feeling is that, when you talk comfort, you are talking about things that are psychological, physiological, and so on—all those deeper ramifications. No one can set up a criterion that will hold in all circumstances. In other words, if you come home and your feet are wet, you want a different temperature in the home than when you come home and you are already nice and warm. All those things are part of the comfort equation and they are hard to express. I really cannot answer what it takes to make comfort. We know, in general, when we are healthy and when we are comfortable and at ease, but there are no words to describe how you can sit down and be comfortable in 72 F.

Beyond this, there are all sorts of boundary conditions that make it difficult to say what you are measuring when you measure something in a house.

1360

No. 1360

TURBULENCE—A FUNDAMENTAL FRONTIER IN AIR DISTRIBUTION

By H. B. NOTTAGE*, Cleveland, Ohio

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

INTRODUCTION AND SCOPE

PROPER air distribution is one of the arts of air conditioning wherein many new data are continually needed to simplify and expedite design work and installation analyses. The general nature of this need in relation to the A.S.H.V.E. Research Program has been discussed previously[1].

This paper, developed under auspices of the Technical Advisory Committee on Air Distribution†, presents an introductory discussion of *turbulence* as an air-distribution phenomenon of vital fundamental importance and is one of a long-term series being developed to meet engineering needs for new data and more comprehensive application principles.

The first simple concepts of turbulence, which are important in appreciating the art of air distribution, will be reviewed. Engineering applications will be indicated for duct friction, turbulent diffusion, and free jets.

THE FORCE-MOTION VIEWPOINT

For better understanding of the fluid-flow problems of controlled air distribution, consider the fundamental question of what goes on within a body of

* Research Associate, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

† TAC on Air Distribution: Ernest Szekely, chairman; N. E. Berry, R. M. Conner, S. H. Downs, F. B. Holgate, F. J. Kurth, J. N. Livermore, R. D. Madison, G. E. McElroy, L. G. Miller, D. W. Nelson, W. A. Pownall, G. B. Priester, T. H. Troller.

Presented at the 55th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January, 1949.

air in motion. A small element of air mass is set into motion by forces acting upon it. Any change in the state of motion is brought about by further force action, with an acceleration or deceleration being opposed by the inertial force of the elementary air mass itself. Thus, an engineering approach to the diagnostic study of air distribution may be built about an *analysis of all of the forces which influence air motion.* Data and design methods recognizing such forces, in relation to the independent variables of an installation, will yield *controlled air distribution.*

The main forces which act within a moving body of air under conventional air-conditioning conditions are: (1) static pressure differences; (2) inertial forces (whenever acceleration acts); (3) buoyancy forces (caused by density differences); (4) shear forces (most evident at solid boundary although also very important within fluid body); (5) gravitational attraction (often a negligible effect in air-conditioning systems).

Given definite expressions for each force entering a particular problem, *equations of motion* may be written which, when solved, yield the velocity and acceleration of the fluid body as a function of position and time. This procedure is easy to state but commonly quite difficult to attain because of practical impediments which are:

1. *Turbulent nature of the air motion:* Where turbulence exists, the pressure, velocity and temperature are in a continual state of rapid, random, fluctuation which can be so complicated as to defy either exact mathematical expression or direct experimental measurement beyond simple exploratory observations.

2. *Inadequate means of expressing turbulence-dependent force in terms of simply attainable experimental data:* In order to describe the fluid behavior in the form of equations which can be solved, all forces and turbulent actions need to be related to the primary easily-measurable variable of the fluid motion. Complicated turbulence behavior must be reduced to a relatively simple form of expression. This can be achieved for important practical cases at present only through experimentally-based empiricisms which lead to approximate engineering solutions.

3. *Mathematical obstacles:* While it may be possible to write the fundamental equations of fluid motion in mathematical notation, it is not always possible to solve these equations and thereby establish concise engineering results. Approximations and empiricisms must then be used and are invaluable engineering tools, although often subject to misuse.

Let it be emphasized, however, that, despite these impediments, there have been established many valuable engineering results based upon the simple fundamentals of turbulence. The future will undoubtedly see many more practical advances, for the research incentive is keen, especially in aerodynamic problems.

ELEMENTARY TURBULENCE CONCEPTS AND QUANTITIES

Turbulence is the physical phenomenon which takes place within a body of fluid in motion whenever small but finite fluid particles undergo an irregular violent commotion and agitation which sets in spontaneously as a process of dissipating (degrading) the available‡ energy of directed fluid motion.

‡ Available energy as used here refers to static pressure and directed kinetic energy.

Manifestation of turbulent flow is observable in the body of a moving fluid through the erratic paths of small suspended particles and the continual random creation, interaction and destruction (as individually distinguishable entities) of innumerable small eddies. In this condition of flow it is not correct to speak of a certain velocity, pressure, or temperature as existing independent of time at a particular point. Instead, each of the variables of the fluid state must be regarded as undergoing continual and irregular *fluctuations.*

Representation of fluctuating quantities is conveniently accomplished by expressing them as the sum of two magnitudes; a *mean* value (mean with respect to time at a fixed position), and a superposed *fluctuating* component. Adopting capital letters for mean quantities, lower-case letters for fluctuating components, and considering a two-dimensional flow as represented on the xy-plane, instantaneous quantities are written as follows:

$$\begin{aligned} \text{Static pressure} &= P + p \\ \text{Temperature} &= T + t \\ \text{Velocity, } x\text{-direction component,} &= U + u \\ \text{Velocity, } y\text{-direction component,} &= V + v \end{aligned}$$

Each fluctuating quantity varies rapidly at random, may be either positive or negative at any instant, and must necessarily have a mean value of zero.

The practical significance of turbulent fluctuations may be demonstrated by considering only the velocity fluctuations in an isothermal constant-static-pressure, incompressible, unidirectional flow stream, *i.e.*, with $V = 0$, $P =$ constant, $\rho =$ constant and $T =$ constant for all positions.

Picture an elementary lattice volume in the stream of unit length perpendicular to the xy-plane, Fig. 1. For conditions of purely laminar flow, when turbulence is not present, the forces per unit area acting in the x-direction are:

1. A static pressure which is the same on both the x and $(x + dx)$ faces.
2. A viscous shear stress which is written as

$$(\tau_v)_y = \mu \left(\frac{dU}{dy}\right)_y$$

or

$$(\tau_v)_{y + dy} = \mu \left(\frac{dU}{dy}\right)_{y + dy} \quad \ldots\ldots\ldots\ldots \quad (1)$$

When turbulence exists, the velocity fluctuations produce *additional stresses,* commonly called the *virtual stresses of turbulence.* For the simple condition in which these turbulence stress components are the same at all points in the region considered, the additional stress magnitudes acting in the x-direction in Fig. 1 are:

1. A pressure caused by the u fluctuations.

$$p = \rho\overline{u^2} \quad \ldots\ldots\ldots\ldots \quad (2)$$

2. A shear stress caused by the transfer of the u-fluctuation momentum in the y-direction through the particle-transporting action of the v fluctuations.

$$\tau_t = -\rho\overline{uv} \quad \ldots\ldots\ldots\ldots \quad (3)$$

NOMENCLATURE†

a_o = pipe radius, feet.

b_o = slot width for Equation 34, feet.

C = concentration, pounds per cubic foot.

c_p = unit heat capacity at constant pressure, Btu per (pound) (Fahrenheit degree).

d = differential operator, dimensionless.

d_o = initial jet diameter for Equation 35, feet.

D = eddy diffusivity, square feet per second.

f = friction factor, dimensionless.

K_1 = empirical constant in Equation 25, dimensionless.

k = thermal conductivity, Btu per (hour) (square foot) per (Fahrenheit degree per foot).

$L = \sqrt{x^2 + y^2}$, feet.

l = mixing length, feet.

l_s = scale of turbulence, Equation 7, feet.

m/A = diffusion mass transfer, pounds per (hour) (square foot).

N_{Re} = Reynolds Number for pipe flow, dimensionless.

P = mean static pressure, pounds per square foot.

p = fluctuating component of static pressure, pounds per square foot.

q/A = heat transfer, Btu per (hour) (square foot).

R = correlation defined by Equation 5, dimensionless.

R_s = correlation defined by Equation 6, dimensionless.

s = time interval, seconds.

T = mean temperature, Fahrenheit degrees.

t = fluctuating component of temperature, Fahrenheit degrees.

U = x-component of mean velocity, feet per second.

u = x-component of fluctuating velocity, feet per second.

V = y-component of mean velocity, feet per second.

v = y-component of fluctuating velocity, feet per second.

U_c = mean velocity on pipe axis, feet per second.

U_{mean} = mean velocity through pipe, feet per second or feet per minute.

U_o = initial mean velocity of jet, feet per second.

$U^* = \sqrt{\tau_o/\rho}$, feet per second.

$U_\tau = \sqrt{\tau/\rho}$, feet per second.

w = source strength, dust particles by weight per second.

x = coordinate dimension, feet.

y = coordinate dimension, feet.

β = eddy conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).

ε = eddy viscosity, square feet per second.

ν = kinematic viscosity, square feet per second.

μ = absolute viscosity, (pound) (second) per square foot.

ρ = mass density, (pound) (second)^2 per (foot)^4.

τ = shear stress, pounds per square foot.

τ_o = shear stress at pipe wall, pounds per square foot.

τ_v = shear stress from viscous action, pounds per square foot.

τ_t = shear stress from turbulent action, pounds per square foot.

† A bar over a symbol or product denotes a mean value thereof.

The bar over a product denotes a mean value and ρ is the fluid density [mass density, (pound) (second)^2/(feet)^4].

The total stresses for turbulent flow are the sum of the turbulent and viscous effects, as indicated in Fig. 1.

The development of the basic ideas continues with the quantities, *intensity, correlation,* and *scale,* and with the condition of *isotropic turbulence.*

1. *Intensity:* Although the mean value of u is zero because on the average, the positive and negative fluctuations cancel each other, the mean square fluctuating velocity, $\overline{u^2}$, is always positive and finite. The mean x-direction kinetic energy of

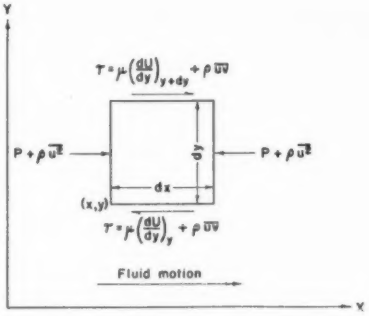


FIG. 1. FORCES PER UNIT AREA ACTING THE x-DIRECTION ON AN ELEMENTARY FLUID VOLUME IN TURBULENCE FLOW WITH T, P, AND ρ CONSTANT AND WITH UNIFORM TURBULENCE CONDITIONS

the turbulent fluctuations is measured by $\overline{u^2}$; and $\overline{u^2}$ thus serves to indicate the *violence* of the turbulence. The intensity of the turbulent velocity fluctuations is defined as

$$\text{Intensity} = \frac{\sqrt{\overline{u^2}}}{U} \quad \ldots\ldots\ldots\ldots\ldots \quad (4)$$

Intensity magnitudes encountered in practice vary from a small fraction of a percent to as much as 20 percent or so.

2. *Correlation:* The condition that the mean product of the two fluctuating velocities at some point in the fluid, $\overline{uv}$, should be other than zero is that a correlation should exist between them. This correlation, R, is a statistical quantity defined as

$$R = \frac{\overline{uv}}{\sqrt{\overline{u^2}}\sqrt{\overline{v^2}}} \quad \ldots\ldots\ldots\ldots\ldots \quad (5)$$

The magnitude of R may vary between 0 and 1, inclusive. If $R = 0$, then the fluctuations u and v are *completely unrelated.* If $R = 1$, then u and v are *uniquely*

interrelated; that is, for each and every value of one of the quantities there is a corresponding single definite value of the other. If R has some intermediate magnitude, it means that for some particular value of one quantity the other quantity will have only a *most probable* value with the chance being that its value may scatter within a considerable range above or below this *most probable* value. The smaller the magnitude of R, the greater is the range of possible scatter.

Fig. 2 illustrates the idea of a correlation. Imagine that *corresponding instantaneous values* of u and v were to be plotted. Sketch (a) shows a condition of no correlation, sketch (b) shows a condition of $R = 1$, and sketch (c) shows a condition corresponding to an intermediate magnitude of R.

3. *Scale:* Specification of a state of turbulence requires some relative measure of the size of the small eddies which characterize the conditions prevailing. Different definitions of scale are possible, and each has its use. The one to be presented here is of particular value in the analysis of turbulent diffusion.

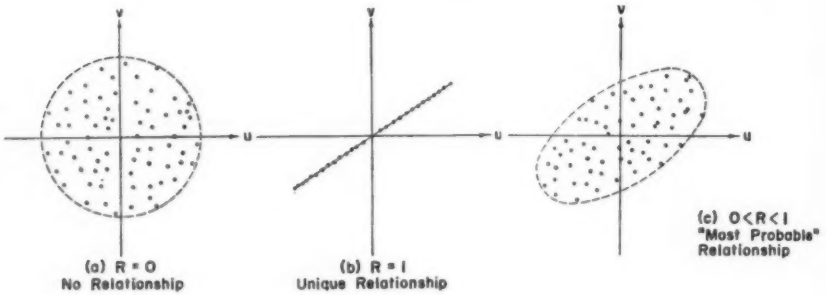


FIG. 2. ILLUSTRATION OF THE CONCEPT OF A CORRELATION BETWEEN THE FLUCTUATING QUANTITIES u AND v

Fluid particles traverse continuous paths in turbulent motion. Imagine some one particle to be followed along its path in a region of uniform mean velocity U and uniform intensity $\frac{\sqrt{\overline{u^2}}}{U}$ and express the extent to which its fluctuating velocity component u_s, existing after a time interval s from some initial instant when $u = u_o$, is dependent upon u_o in terms of the correlation R_s defined as

$$R_s = \frac{\overline{u_o u_s}}{\overline{u^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

where

$$\overline{u^2} = \overline{u_o^2} = \overline{u_s^2} \text{ by the postulate of uniform intensity.}$$

At the initial instant, of course, $R_s = 1$; and as the time interval grows, R_s approaches its ultimate magnitude of zero. When $R_s = 0$ it means that the particle's fluctuating velocity then is no longer in any way dependent upon u_o. Since the motions of all particles which pass through a turbulence eddy or whirl must have some sort of characteristic continuity, one possible *scale* of the turbulence would be a proba-

bility-weighted mean distance of particle travel during the time interval required for R_s to become zero. Thus, define the scale l_s as

$$l_s = \sqrt{\overline{u^2}} \int_0^{s_0} R_s \, ds \quad \ldots \ldots \ldots \ldots \ldots \quad (7)$$

Where the upper limit s_0 is the time interval required for R_s to become zero (and for $s > s_0$, $R_s = 0$ at all times). An equivalent expression is then

$$l_s = \sqrt{\overline{u^2}} \int_0^{\infty} R_s \, ds \quad \ldots \ldots \ldots \ldots \ldots \quad (8)$$

Scale magnitudes are usually less than an inch in engineering problems, although a phenomenon such as atmospheric turbulence may have a quite large scale.

4. *Isotropic Turbulence:* In the most general state of turbulence the scales and intensities have different magnitudes in different directions at any point in the fluid and are further different between any two points in the same direction. This case is at present too complicated to permit mathematical analysis to obtain practical results, and even an experimental study of such fluctuations involves a tremendous, almost prohibitive amount of instrumentation detail. By far, the greatest amount of factual knowledge and analyses to date have been established for the simplest condition of turbulence, called *isotropic* turbulence.

In isotropic turbulence, the intensity and scales have the same magnitudes in all directions at any one point; hence, a single specification or a single measurement of these quantities completely determines the characteristics of the turbulence at the point being considered. Strict isotropy requires (as a consequence of the defining conditions) that the correlation R, Equation 5, be zero, which means that the turbulent shear stress of Fig. 1 must be zero and that all fluctuations are completely at random.

Turbulence in engineering flow systems, strictly speaking, is almost always nonisotropic; but nonisotropic turbulence has a strong natural tendency to become isotropic if interferences such as solid boundaries, high gradients of pressure, obstacles, and like effects are minimized or far removed from the location being considered. Hence, as a practical approximation, the theories of isotropic turbulence have been extended to the utmost for engineering expediency.[2]

SOME ENGINEERING EMPIRICISMS

Among the many empiricisms which have been variously considered to simplify turbulence as a *practical* problem, only a few of the more familiar ones will be mentioned here. Despite fundamental weaknesses, these have been of engineering utility. To be considered are the *eddy viscosity, eddy conductivity, eddy diffusivity,* and a kind of scale known as the *mixing length.*

1. *Eddy viscosity:* While it is a fundamental accomplishment to express the virtual shear stress due to turbulence (Fig. 1) as

$$\tau_t = -\rho \, \overline{uv},$$

engineering problems cannot be solved until the product $\overline{uv}$ is somehow expressed in terms of the independent variables of the flow. An early proposal for doing this was to assume that turbulence had an effect similar to an increase in the magnitude of the absolute viscosity, μ. Thus, an *eddy viscosity,* ϵ, was invented which was to satisfy the defining equation,

$$\tau t = \rho \varepsilon \frac{dU}{dy} \quad \ldots \ldots \ldots \ldots \ldots \quad (9)$$

The total shear stress then was expressed in the form

$$\frac{\tau}{\rho} = (\nu + \varepsilon)\frac{dU}{dy} \quad \ldots\ldots\ldots\ldots\ldots \quad (10)$$

where

$\nu = \mu/\rho$, the kinematic viscosity of the fluid.

Extensive experimental experience was needed to validate this step. If ε proved to be a definite characteristic of each type of flow and to be a simple function of the independent variables involved, then it could be accepted as a sort of *property* of turbulent flow. Accumulated evidence has served to disprove this broad hope, however, and the eddy viscosity has become an empiricism which may be convenient but can be used only where data have been established to predetermine its magnitude.[3]

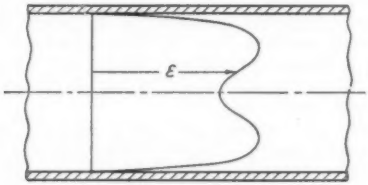


FIG. 3. FORM OF THE EDDY-VISCOSITY VARIATION ACROSS A CIRCULAR PIPE IN FULLY-DEVELOPED TURBULENT FLOW

The typical variation of eddy viscosity across a pipe diameter for fully developed turbulent flow is sketched in Fig. 3. The maxima there shown at a position slightly less than half a pipe radius out from the wall indicate the region of most violent turbulent commotion and eddy-mixing action. Recognition of this is important when analyzing turbulent heat or mass transfer in pipes, for example.

2. *Eddy conductivity and eddy diffusivity:* The virtual shear stresses of turbulence arise as a consequence of the turbulent transfer of fluid-particle momentum across surfaces within a fluid body. In a similar manner, turbulence has a strong influence upon heat transfer or mass transfer (diffusion) within a fluid body, and the empiricism of the eddy viscosity has been paralleled by the invention of *eddy* quantities for turbulent heat and mass transfer.

Consider a laminar fluid flow in the x-direction. The rate of heat transfer across the stream, in the y-direction, may be written as

$$\left(\frac{q}{A}\right)_v = -k\frac{dT}{dy}, \text{ Btu per (hr) (sq ft)} \quad \ldots\ldots\ldots \quad (11)$$

where

k = the thermal conductivity and (dT/dy) is the temperature gradient.

Next consider that the stream becomes turbulent and define an *eddy conductivity*, β, by the relation.

$$\left(\frac{q}{A}\right) = -(k + \beta)\frac{dT}{dy}, \text{ Btu per (hr) (sq ft)} \quad \ldots\ldots\ldots \quad (12)$$

For mass transfer, imagine that some substance is diffusing across the laminar stream. The rate of diffusing mass flow is

$$\left(\frac{m}{A}\right)_v = -\eta \frac{dC}{dy}, \text{ lb per (hr) (sq ft)} \quad \ldots\ldots\ldots \quad (13)$$

where η is the diffusivity for the substance and the fluid concerned, and (dC/dy) is the concentration gradient for the diffusing substance (concentration is expressed as the pounds of diffusing substance per cubic foot of fluid.)

In turbulent flow, define an *eddy diffusivity, D,* by the relation

$$\left(\frac{m}{A}\right) = -(\eta + D) \frac{dC}{dy}, \text{ lb per (hr) (sq ft)} \quad \ldots \quad (14)$$

Generally in fully-developed turbulence the viscous terms are negligibly small in comparison to the *eddy quantities* in Equations 10, 12, and 14, so that the turbulent processes can be described directly as:

Momentum transfer,

$$\tau/\rho = \varepsilon \frac{dU}{dy} \quad \ldots\ldots\ldots\ldots\ldots \quad (15)$$

Heat transfer,

$$q/A = -\bar{\rho} \frac{dT}{dy} \quad \ldots\ldots\ldots\ldots\ldots \quad (16)$$

Mass transfer,

$$m/A = -D \frac{dC}{dy} \quad \ldots\ldots\ldots\ldots\ldots \quad (17)$$

3. *Analogy approximation interrelating the eddy quantities:* Engineering problems are frequently expedited, where data are otherwise lacking, by the use of some sort of an analogy between momentum, heat, and/or mass transfer. If the turbulence mechanisms of these three phenomena were similar, which is in many simple cases a workable, although inexact, approximation to be assumed, then Equations 15, 16 and 17 could be further interrelated. The assumption would be expressed as

$$\varepsilon \cong \frac{\bar{\rho}}{3600\rho\, c_p\, g} \cong D, \text{ sq ft/sec} \quad \ldots\ldots\ldots \quad (18)$$

where

$g = 32.2$ ft/(sec)^2 and 3600 is a time-unit conversion, hours to seconds.

Successful practical applications of Equation 18 have been made.[4]

4. *The mixing length:* An early turbulence empiricism which received considerable attention for a time was the so-called *mixing length.* A simple development of this idea will be given, following Prandtl.[5]

Consider again a turbulent flow parallel to the x-axis with the mean velocity U a function of y only. The shearing stress on a horizontal plane in the direction of U, neglecting ν compared to ϵ, is of magnitude

$$\tau = \rho\, \overline{uv} = \rho\, \varepsilon \frac{dU}{dy} \quad \ldots\ldots\ldots\ldots\ldots \quad (19)$$

This shows that (τ/ρ) is dimensionally the square of a velocity; let this velocity be

$$U_s = \sqrt{\frac{\tau}{\rho}} = \sqrt{\overline{uv}} \quad \ldots\ldots\ldots\ldots\ldots \quad (20)$$

(where positive magnitudes are always understood under the square root signs). The eddy viscosity ϵ has the dimensions (length + velocity). Assume that U_* is this latter velocity, for it is the most significant one directly associated with the turbulence, and introduce an arbitrary length l to write

$$\epsilon = l\, U_* \quad \ldots \quad (21)$$

Substituting Equation 21 and the relation $\tau = \rho\, U_*^2$ into Equation 19, there follows

$$U_* = l \frac{dU}{dy} \quad \ldots \quad (22)$$

The eddy viscosity can alternatively be expressed

$$\epsilon = l^2 \frac{dU}{dy} \quad \ldots \quad (23)$$

Equation 19 for the shear stress then can be written††

$$\tau = \rho l^2 \left(\frac{dU}{dy}\right)^2 \quad \ldots \quad (24)$$

The quantity l is called the *mixing length;* its advantage is merely its simple definition, and it is not a truly fundamental quantity. Its use may be empirically helpful for simple flow systems wherein both the shear stress and the mean-velocity distribution are known to follow simple formulas, for then l will be some simple function of position and of the flow variables. Wherever l is known in simple form, solution of the equations of motion for turbulent flow is facilitated.

The name *mixing length* arose from another method of derivation wherein turbulent fluid entities are pictured as producing a shear stress by *shooting* back and forth between adjacent *layers* of fluid; and the mean distance of such travel is then l. From this latter aspect, l is a sort of *scale* of the turbulence.

BRIEF APPLICATION EXAMPLES

Friction in Ducts and Pipes: Currently recommended duct-friction factors[6] were established from a systematic empirical treatment of turbulent-flow data. The development will be briefed here for smooth round ducts.

First solve Equation 24 to obtain an expression for the mean velocity, U, as a function of radial position. Let y denote the distance from the duct wall measured radially *inward*. Then:

1. By dimensional reasoning, set the mixing length proportional to some dimension characterizing the turbulence. The distance y is the simplest one, so take

$$l = K_1\, y \quad \ldots \quad (25)$$

2. In pipe flow the shear stress, τ, is zero on the axis and increases linearly along the radius to reach a maximum, τ_0, at the wall; *i.e.*,

$$\frac{\tau}{\tau_0} = 1 - \frac{y}{a_0} \quad \ldots \quad (26)$$

---

†† Observing that the sign τ must change as the sign of (dU/dy) changes, a mathematically rigorous expression is $\tau = \rho l^2 \left|\frac{dU}{dy}\right| \left(\frac{dU}{dy}\right)$

where

a_o = the pipe radius.

3. Upon substituting Equations 25 and 26 into 24 and integrating to obtain a relation between U and y, there is obtained a velocity-distribution equation which can be tested by experimental data.[5, 7] It so happens, however, serving as a demonstration of the peculiarities of empiricisms, that the result of these steps is in poorer agreement with experiment than is the result of assuming that $\tau = \tau_o$ throughout the pipe and then

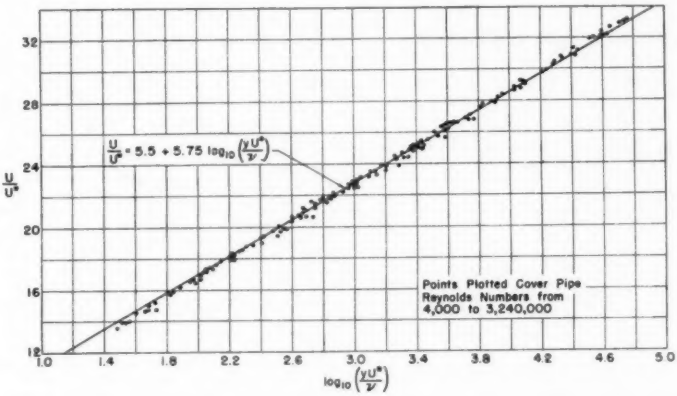


FIG. 4. NIKURADSE'S DATA ON THE VELOCITY DISTRIBUTION FOR TURBULENT FLOW IN SMOOTH PIPES AS EMPLOYED TO ESTABLISH THE VELOCITY-DISTRIBUTION FUNCTION OF EQUATION 29

solving for the velocity distribution. This latter procedure yields Prandtl's famous logarithmic velocity-distribution law,

$$\frac{U_c - U}{U^*} = \frac{1}{K_1} ln_e \left(\frac{a_o}{y}\right) \quad \ldots\ldots\ldots\ldots \quad (27)$$

where

$$U^* = \sqrt{\frac{\tau_o}{\rho}}$$

U_c = mean velocity at the pipe axis

This form of expression agrees very well with the velocity-distribution data of Nikuradse,[5, 7] although its *derivation* is not fundamentally rigorous. But any equation which represents engineering data over an established range has practical value no matter how it may be obtained, and proceeding from the demonstrated utility of Equation 27 further useful relationships are attainable. Note that U^* and the conventional friction factor,[6] f, are related:

$$U^* = \sqrt{\frac{\tau_o}{\rho}} = U_{\text{mean}} \sqrt{\frac{f}{8}} \quad \ldots\ldots\ldots\ldots \quad (28)$$

where

U_{max} = mean fluid velocity over the pipe cross section.

Final equations of *great practical importance,* with empirical constants determined from Nikuradse's data for turbulent flow in smooth pipes, are the following:

The *velocity distribution:*

$$\frac{U}{U^*} = 5.5 + 5.75 \log_{10}\left(\frac{yU^*}{\nu}\right) \quad \ldots\ldots\ldots\ldots \quad (29)$$

The *friction-factor relationship:*

$$\frac{1}{\sqrt{f}} = 2 \log_{10} (N_{Re} \sqrt{f}) - 0.8 \quad \ldots\ldots\ldots\ldots \quad (30)$$

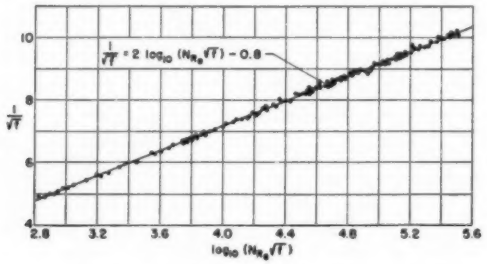


FIG. 5. NIKURADSE'S DATA ON FRICTION FACTORS FOR TURBULENT FLOW IN SMOOTH PIPES AS EMPLOYED TO ESTABLISH THE FUNCTION OF EQUATION 30

Figs. 4 and 5 show how Equations 29 and 30 represent Nikuradse's data. The velocity distribution§ refers only to the turbulent *core;* near the wall, viscous effects become important and the flow pattern changes. Other similar relationships are applicable to rough pipes[7].

The preceding material is but one example of the many ways in which turbulence empiricisms have been of great practical utility to the engineer in treating turbulent flow.

Turbulent Diffusion: The processes of turbulent diffusion whereby mass or heat spreads across a turbulent stream are of wide technical importance. Turbulent diffusion in substantially isotropic streams, of cross section large enough to permit instrumentation without distorting the flow, can be studied by relatively simple methods; and this fact has been used in obtaining measurements of intensity and scale for application to various practical problems[8, 9, 10].

One little-investigated application to an air-conditioning problem would be the treatment of dust transport and deposit in large ducts according to the theory

§ The validity of the Nikuradse data, in the immediate vicinity of the pipe wall, has recently been questioned, but the region in question has been excluded from the range of Fig. 5.

of sediment transport in turbulent streams[11]. A very simple related problem, suitable as a brief illustration, would be the spread of dust particles introduced in the air stream of a large duct at a uniform rate from a *point source.* The equation of turbulent diffusion will be given for this case, neglecting any gravitational settling of particles, and applicable under conditions where: (1) the mean velocity U is constant, (2) the turbulence is at least approximately isotropic, (3) the intensity and scale of the turbulence are constant (or, alter-

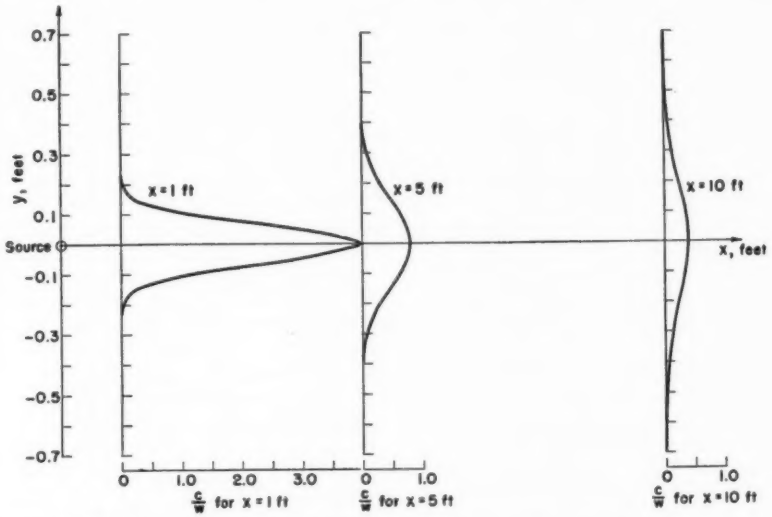


FIG. 6. RELATIVE DISTRIBUTION OF DUST-PARTICLE CONCENTRATION DOWNSTREAM FROM A POINT SOURCE (EQUATION 33)

natively, the eddy diffusivity is constant), (4) the dust particles have a small enough inertia not to be thrown out of the turbulence eddies, and (5) the distance downstream from the source is greater than l_s. The dust particle concentration[12] as a function of the position coordinates (x, y), with the source at the origin, and letting

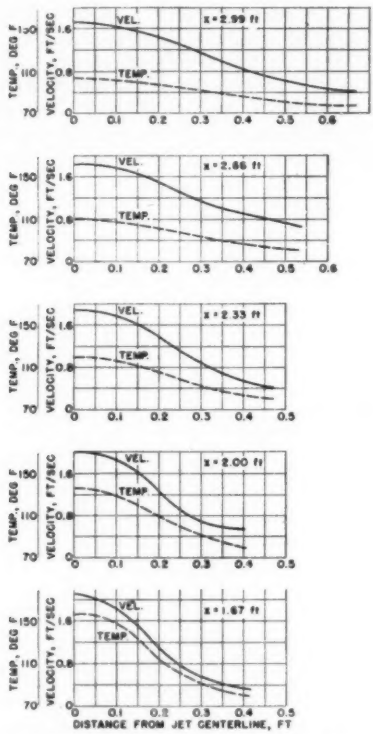
$$L = \sqrt{x^2 + y^2}, \text{ is } C = \frac{w}{4\pi DL} e^{\frac{-U}{2D}(L - x)} \quad \ldots\ldots\ldots (31)$$

where

C = dust particle concentration in parts by weight per unit volume.
w = the source emission, dust particle parts by weight per second.
D = eddy diffusivity, square feet per second.

When $(L - x)$ is a small quantity, the sagitta formula gives

$$L - x \cong \frac{y^2}{2x} \quad \ldots\ldots\ldots\ldots\ldots (32)$$



x=distance referred to equivalent point source of jet. Surrounding air at rest with temperature of 71.1 F. Jet rising vertically.

FIG. 7. EXPERIMENTAL VELOCITY AND TEMPERATURE PROFILES FOR A HEATED CYLINDRICAL AIR JET (TAKEN FROM SCHMIDT[17])

A corresponding form of Equation 31 is

$$\frac{C}{w} = \frac{1}{4\pi D\left(x + \frac{y^2}{2x}\right)} e^{-\frac{U y^2}{4Dx}} \quad \ldots\ldots\ldots\ldots (33)$$

Fig. 6 demonstrates Equation 33 for $U = 10$ fps and $D = 0.02$ sq ft per second.

Phenomena of turbulent diffusion are both abundant and important; smoke dissipation, exhaling of breath, sediment transport in streams, and the dispersion of water-droplet mists are examples.

Free Jets: So long as workable formulations of turbulence phenomena require empiricisms, the analysis of turbulent jet behavior will also require some sort of empiricism. Some of the component features of jet behavior are predictable, however, on the basis of present knowledge. For example, theory[13] predicts the course of the centerline jet velocity as follows:

1. For a two-dimensional isothermal plane jet (*e.g.*, jet from very long and narrow slot), when a condition of fully-developed turbulence has been reached the centerline velocity should fall off in proportion to the inverse of the square root of the distance from the source.

2. For a cylindrical isothermal jet, when a condition of fully-developed turbulence has been reached the centerline velocity should fall off in proportion to the inverse of the distance from the source.

These predictions were checked against experimental data by Tuve and Priester[14]. For discharge into a large free space condition 1 was not attained for jets from long narrow slots with aspect ratios as high as 128:1. Slots from the rectangular outlets tested all tended towards a circular cross section at sufficient distances from the outlet. These authors also found that condition 2 was only approximately followed by their data; instead of direct proportionality being maintained to the end of the jet, the observed proportionality constant decreased in magnitude as the end of the jet was approached. Interpretation can only be variously surmised, but the basic phenomenon to be better understood is *turbulence.*

The recent data of Albertson et al[15], on the other hand, show good agreement with both conditions 1 and 2. Two-dimensional plane jets were formed from slots of aspect ratios 288:1 and greater, with the jets discharging between two parallel walls set perpendicular to the plane of the slot outlet and perpendicular to the long side of the rectangular slot at the slot ends. This condition typifies a situation occasionally met in air-conditioning practice. The equation given for the two-dimensional jet (with rounded approach to the slot) was

$$\frac{U_{max}}{U_o} = 2.28\sqrt{\frac{b_o}{x}}, \text{ for } \frac{x}{b_o} > 6, \quad \ldots\ldots\ldots\ldots \quad (34)$$

where

U_{max} = the centerline jet velocity,
U_o = the initial jet velocity,
b_o = the slot width in feet,
x = the normal distance from the outlet in feet.

Equation 34 was established for initial velocities from 2,400 to 10,800 fpm, and the lowest jet velocity measured was 100 fpm.

The equation given by Albertson et al[15] for cylindrical jets was

$$\frac{U_{max}}{U_o} = 6.2\,\frac{d_o}{x}, \text{ for } \frac{x}{d_o} > 7, \quad \ldots\ldots\ldots\ldots \quad (35)$$

where

d_o = the initial jet diameter.

Equation 35 is in substantial agreement with the results of Tuve and Priester[14] and thereby lends support to their more extensive data.

Equations 34 and 35 are both limited to isothermal conditions.

The non-isothermal jet is very inadequately understood[16], despite its great practical importance. The nature of the unsolved problems will be indicated here by presenting the data of Schmidt[17]. Fig. 7 shows his cross-stream temperature and velocity traverses for a cylindrical jet at various distances from the origin. The evident dissimilarity between the two profiles complicates any analysis, and yet both temperature and velocity are of air-conditioning importance.

The analysis of a heated cylindrical air jet rising vertically as developed by Schmidt further showed the centerline velocity to vary as $x^{-1/3}$ and the centerline temperature as $x^{-5/3}$, where x is the distance along the jet measured from an equivalent point source. The equivalent point source for a finite outlet may be established by experiment or approximately located upstream from the outlet plane at the position of the vertex of the jet expansion angle. Much further study is needed on all of these matters.

CONCLUSIONS

1. The true, microscopic, fundamental mechanisms of turbulence are at present incompletely understood.

2. Carefully-conceived engineering empiricisms offer the best means for obtaining workable solutions to turbulence problems of practical importance. Such empiricisms must be guided by the best analytical work attainable, the total of past experience, and the results of very careful experimentation.

3. Much additional experimental research is needed on engineering problems of turbulence flow and air distribution, and improved instrumentation in this connection is a prime requirement.

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DISCUSSION

G. L. TUVE, Cleveland, Ohio: In calling for discussion I would like to point out one or two things. This paper is a real accomplishment in our research program. It marks an outstanding effort to translate or show the transition from the old *test this and test that and see if it works* method to a newer method which has been introduced largely by the German aerodynamacists—*Write equations, then interpret and apply.*

One of the difficulties encountered by Mr. Nottage, or anyone who attempts to use this newer method, is the matter of language. We cannot understand each other. You may have run into something you could not understand in the paper. The aerodynamacist and the fluid-mechanics specialist tend to use what I call a mixture of German and Greek. They use Greek symbols and translated German words. Our basic material in this field has come largely from what grew up at Gottingen and other German centers, and our text books on these subjects are often literal translations from the German texts. Those of us who are busy trying to tie together the practice of heating and ventilating engineering and the basic fundamentals of fluid mechanics, are having a hard time because the two groups of people who are expert in these fields talk different languages.

I wanted to make that remark because I think Mr. Nottage has done a pioneering job in making this translation for us. It is at least a start. Maybe there are some more translaters in the audience.

CYRIL TASKER, Cleveland, Ohio: In the talk that I gave the other day, I quoted from the director of the Building Research Board, who said that our research work should not only find the *what* but the *why*. This is one of the projects in which we are trying to find the *why*, just as much as the *what*.

Mr. Nottage has shown you a picture of the experimental setup which has been erected at the Laboratory for the study of ventilation jets and that work is actually under way. In the process of finding the *what* at the Laboratory, Mr. Nottage, with his background and experience, is at the same time attempting to find the *why*. When we put the two of them together, I think we shall come up with answers that will be of real value to the members of the Society.

As Professor Tuve has said, it is a tough problem and I think that we are fortunate at the Laboratory in having Mr. Nottage, who can bridge that gap between the German experimenters and experimental work in this country and in other countries. We are definitely trying to give you practical data that will be of use in air distribution. But I think there is a great deal of work that has been done over the past years by various investigators and you still do not exactly know what happens when you throw a jet out of an outlet. I think that illustrates the need for some experimental work of this type and I hope that we are going to be permitted to carry on with this until we can give you some really valuable, practical results.

M. K. FAHNESTOCK, Urbana, Ill.: This is an analytical type of paper and the remarks which I have to make pertain to the need for more work of this type and the need for agreement on the analytical approach to the problems of air distribution before actual laboratory research work is undertaken. As engineers, some may question, *why should the Society be doing, or supporting, analytical work?* I believe that reference to the Society's Constitution, By-Laws and Rules pertaining to its research activities will show that both theoretical and applied research are covered and encouraged.

Much of our past research has been applied, or empirical. This is probably rightly so because we are a society of engineers and many of our problems of common concern can be answered by applied research. Many times the results from this type of research are limited to the test conditions and too often are not entirely applicable to the wide range of conditions experienced in practical applications. Fundamental research of an analytical character usually leads to the determination of coefficients or constants in equations arrived at from an analytical or mathematical treatment of the factors or variables involved. The final results, usually in the form of mathematical equations, should, and usually do, have wider applications than the results from applied research.

I believe that the Society should conduct and support both types of research, with the emphasis reasonably placed to best serve the needs of the members. I suggest that in the future more emphasis might well be placed on basic research. Furthermore, I suggest, in the planning of basic research, where much of the work is analytical in character, that there be co-ordination and agreement among at least several analysts on a specific problem before too much time is spent on it, or the results published.

I am sure that Mr. Nottage would appreciate the opportunity of discussing his ideas of certain analytical approaches to certain air distribution problems with others who are interested and qualified to approach the problems analytically. In fact, he has probably already done so. I think we should encourage him and others who are willing and capable of approaching our problems analytically and giving us results in engineering terms which we can understand and use. Therefore, I suggest that consideration be given to the thought of bringing together at least three or four people, with the training, background, and interest which qualify them to approach our air distribution problems analytically and to present the answers so

they can and will be used by engineers. No doubt, Mr. Nottage can recommend some individuals with the necessary qualifications and interests and with whom he would like to work.

Agreement on the analytical theory by a group of qualified analysts would give greater assurance that the theory was correct and acceptable. It would give insurance that it would withstand future scrutiny. Such a group of individuals could well be named as a subcommittee of the present TAC on Air Distribution.

JOHN AEBERLY, Chicago, Ill.: This type of research is going to carry us and the Society farther in the long run than much of the research work done in the past. There are two thoughts which should be carefully considered, and, when these are properly understood, they will help materially to solve our problems.

First, scientists invariably use an interim language to bridge the gap between the formulation of hypotheses and the establishment of the final physical facts. The members of this Society have had considerable experience with this language in our frequent dealings with medical science. This terminology has been difficult for us to understand and at times we have actually been bogged down by it. The involved language in this type of paper does not enable us to comprehend clearly and quickly the important facts presented.

Underlying the material contained in this important paper are two fundamental physical laws. It would seem that the author and others who will deal with this problem should avoid using interim expressions and correlate their findings with the fundamental laws of conservation of energy and conservation of momentum.

Secondly, this paper presents important data treating with matter in its molecular state. All of these particles are resilient; in fact, for all practical purposes of our membership, it may be presumed that these particles have perfect resilience. Why, then, can we not express our findings through the use of words dealing with energy and momentum exchange of perfect resilient bodies?

A recent experience in an attempt to establish the percentage of short circuiting which takes place under certain conditions in a supply and return register should serve to illustrate this simplification. The research workers, the men of this industry who had a thorough understanding of our basic knowledge, and others made assumptions regarding heat exchange, mass movement, and the action of gases other than air which led to decidedly erroneous findings. It was only after we considered our problems in connection with these two basic physical laws, and agreed upon certain approximations, that we were able to obtain values consistent with the experimental results. The divergence of opinions was so marked that I really feel this Society should long ago have undertaken exactly the type of research contained in the preparation of this paper. If we can simplify our entire procedure by basing it upon these two indisputable physical laws, we can avoid extravagance in our research program.

W. O. HUEBNER, New York, N. Y.: Many of us who have approached the air distribution problems from the practical point of view, and have sometimes approached them from the theoretical point of view, are happy about the work done by the research staff and especially for the work done by Mr. Nottage.

AUTHOR'S CLOSURE: In closing the discussion, it is well to point out that the oral presentation of this paper was an intentional departure from the written material. The presentation gave emphasis to the A.S.H.V.E. Research Laboratory's air-distribution research program and the plans to make the utmost use of recent advances in practical turbulence knowledge. The encouragement given by the discussers is most helpful.

The material presented has not been created as original work at the Laboratory, but the plans to use this information so as to save research time and effort in establishing engineering means to improve air distribution are the new approaches to old problems as recommended by the Technical Advisory Committee on Air Distribution. The paper itself is a highly abridged introduction to a comprehensive survey of the background literature on turbulence and ventilation jets in air distribution which is currently being readied for release at the Research Laboratory.

A special subcommittee has been formed under the TAC on Air Distribution to further discuss analytical and experimental details as the program develops. Use of the fundamental laws of fluid flow is the only basis for the proper planning and interpretation of experimental studies. The entire program at the A.S.H.V.E. Research Laboratory is being developed about this principle.

1361

No. 1361

BALANCING AIR DELIVERY OF A SYSTEM OF MANIFOLD AIR DIFFUSERS

By GEORGE S. DAUPHINEE* AND PETER ARGENTIERI†, NEW YORK, N. Y.

THE final adjustment and regulation of the air delivery through multiple supply air diffusers or outlets in a ventilating system for the purpose of bringing their individual air discharge volumes in conformity with those specified is commonly known as *balancing* the system. At present, the method generally adopted is one of trial and error, and is an admittedly costly and unscientific operation.

Regardless of the method employed for balancing, the precision of adjustment will be influenced by the original design and construction of the system, such as the relative proportioning of the distributing ducts and the type and construction of take-offs, air turns and dampers. It is, for example, conceivable that the system design and installation could be so adverse that final regulation to produce the desired or prescribed air distribution would be impossible.

The purpose of this paper is two-fold: (*a*) to outline a simple and positive method for balancing multiple air diffusers in a common ventilating system which would also lend itself to standardization, and (*b*) to emphasize the vital importance of proper duct design to insure easy balancing of the system after it is installed.

It is impracticable to so design a duct system that the exact required air volume will be discharged from each respective outlet automatically and without the aid of adjustable dampers. It is further recognized that, unless such dampers are calibrated *functionally, i.e.,* to produce a known result directly, tedious trial and error in balancing will be unavoidable. This paper precludes the use of dampers in the ducts to regulate the air delivery through individual outlets and requires the employment of dampers which are part of the outlet and have a known effect on outlet performance.

A system of air outlets is balanced when (*a*) each outlet discharges its specified *percentage* of the total air delivery and (*b*) such total air delivery

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conforms to that specified. The balancing method to be described consists, therefore, of two steps performed in the following sequence: (1) adjusting the relative air delivery from each air outlet or diffuser *percentage-wise* (without regard to *actual* air delivery) and, (2) checking, and, if necessary, adjusting the fan speed so that the proper total actual air delivery is obtained.

To accomplish step 1, the following operating data for each diffuser are required:

(*a*) Capacity factors for determining the volumetric air delivery of any size diffuser from its air outlet velocity.

(*b*) Calibrated damper factors for establishing the damper adjustment of any outlet to produce any desired percentage of its *full open* capacity.

BALANCING PROCEDURE

Given a typical installed ventilating system with a multiplicity of air diffusing outlets (diffusers), all dampers should first be fully opened and capacity readings taken for each outlet. For this operation, it is not necessary that the fan volume

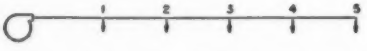

FIG. 1. AIR DELIVERY SYSTEM FOR EXAMPLE 1

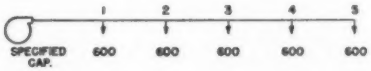

FIG. 2. AIR DELIVERY SYSTEM FOR EXAMPLE 2

conform to specifications, but it is important that it be maintained constant while all readings are made. Next, a calculation should be made to determine the ratio of actual air delivery of each diffuser to the total actual air delivery from all the diffusers. By comparing these ratios with those required by the specifications, the degree of variation in the air *apportionment* among the outlets from that desired will be established and will serve as the basis for damper adjustment of individual outlets. The percentage of damper opening required for each diffuser in order to regain the apportionment specified is then calculated, and, using the damper adjustment factors, all dampers (but one—see examples) are set accordingly. After dampers have been set, an air delivery determined for any one diffuser will indicate whether its air delivery is in accordance with, or varies from, that specified for the respective diffuser. If at variance, it will be known that the capacity of all outlets varies from their specified capacities *in the same degree* and, then, by adjusting the fan speed to produce the capacity required for any one outlet, all other outlets will deliver their respective specified capacities. A final check of several outlets will serve to prove the accuracy of balance.

It is emphasized that the procedure outlined is far simpler than may at first appear, since most of its operations are slide rule computations which can further be facilitated by preparatory tabulation. The manual contribution consists entirely of (1) opening all dampers, (2) determining air delivery from each diffuser, (3) setting dampers and (4) checking the output of several outlets. The method will be clarified by the typical examples which follow. For purposes of simplification only, these examples are arbitrarily limited to five

outlets. It should be appreciated, that the method and procedure are adaptable to systems with any number of outlets supplied by a common fan as demonstrated by the concluding example showing the results of an actual large-scale balancing operation.

The diffusers used in all examples contain a built-in sleeve type damper, the function of which is to vary the outlet aperture. It is raised or lowered by a central screw operated from below the diffuser. The manufacturer's damper adjustment factors are used for obtaining any desired percentage of the *full open* damper capacity converted to number of screw turns from the full open damper position. These damper factors, therefore, correspond to the ratio of *actual restricted* to *actual full open* air delivery. With this particular type damper, the air delivery will be:

$$(\text{cfm}) = V \times F \times \frac{c}{100} \quad \ldots\ldots\ldots\ldots \quad (1)$$

where

V = outlet velocity, fpm.
F = capacity factor.
c = percent of full open damper capacity.

The outlet, of course, could be so designed that the normal position of the damper is in midway or partially open position, permitting adjustment in either the closing or opening direction with corresponding calibration factors. For certain types of dampers, the velocity value will vary directly as the capacity, and the air delivery will be simply:

$$(\text{cfm}) = V \times F \quad \ldots\ldots\ldots\ldots \quad (2)$$

Air outlet velocities are determined at the periphery of the diffuser with a standard deflecting vane type anemometer, four readings being taken at points 90 deg apart and averaged. Readings to the nearest 25 fpm are considered sufficiently accurate. The anemometer readings are multiplied by the capacity factor previously established to convert the reading to equivalent air delivery in cubic feet per minute.

Example 1. An air delivery system shown in Fig. 1 has 5 outlets for which the specified air delivery in cubic feet is as follows: No. 1—800, No. 2—800, No. 3—1230, No. 4—1100 and No. 5—560. The total air volume to be delivered is 4490 cfm. Outline procedure for balancing the system.

Solution: The various steps to be taken are given in Table 1.

The tabulation in Table 1 has been purposely elaborated to illustrate all operations. For example, Lines c and j are inserted merely to exemplify the establishment of the specified air apportionment because this is, actually, the basis of the method. The function of the values in line G is to establish the outlet to be used as a *base;* the outlet having the lowest percentage of actual delivery volume. This often can be determined merely by inspection without computation of other percentages on this line. Likewise, the values in Line J, while representing the relative air deliveries which would prevail after balancing *if* the observed actual air delivery of the *base* outlet remained constant, are hypothetical, since the effect of balancing alters the air delivery of the *base* diffuser. In practice, therefore, the computation of line J is omitted and the

percentage of full open damper delivery (line K) obtained directly by dividing values in line H × 100 by values in line G. It is further emphasized that, in performing the balancing operation, the tabulation is prepared in advance (omitting lines c, J and j) and the data in lines A, B and C are entered thereon from the plans and specifications. Deflecting vane type anemometer readings are then taken for all diffusers at once, entered in line D, and the values in lines F to K inclusive computed. Following this, the dampers are set, likewise at once, in accordance with line K.

It will be observed that, as all dampers are in the open position when determining actual air deliveries, all adjustments must necessarily be in the throttling direction and that, accordingly, the damper in the *base*, or outlet least favored, will always remain fully open, all other diffusers being throttled to the extent necessary to apportion the total air delivery among all diffusers in accordance with specified requirements. It will be observed further that this is accomplished without regard to, and independent of, the actual fan delivery.

TABLE 1. BALANCING OF AIR DELIVERY FOR *Example 1*

A Diffuser Location	1	2	3	4	5	Totals
B Diffuser Size (Inches Neck Dia.)	12	12	14	14	10	
C Air Volume Specified, cfm	800	800	1230	1100	560	4490
c Percent of Total Air Specified	17.8	17.8	27.4	24.5	12.5	100
D Anemometer Reading (fpm) with All Dampers in Full Open Position	2025	1950	1875	1800	1750	
E Manufacturer's Capacity Factor	0.45	0.45	0.61	0.61	0.31	
F Actual Air Delivery (cfm) = line E × line D	910	877	1143	1098	543	4571
G Variation from Specified = line F/line C	1.14	1.10	0.93	1.0	0.97	
H *Base* Diffuser = Least Value in line G			0.93			
J cfm Required for Specified Ratio Referred to *Base* = line H × line C	744	744	1143	1022	521	4174
j Percent of Total line J = line c	17.8	17.8	27.4	24.5	12.5	100
K Percent of Full Open Damper Delivery Required = $\frac{\text{line J} \times 100}{\text{line F}} = \frac{\text{line H} \times 100}{\text{line G}}$	81.7	84.8	100	93	96	

After the dampers have been adjusted, an anemometer recheck of the *base* diffuser will indicate whether it delivers the required capacity. If not, it will be known that the air delivery of every diffuser of the system varies from the required capacity *in the same ratio* and will be corrected automatically by adjusting the fan delivery accordingly. In *Example 1*, for instance, the velocity reading for the *base* diffuser (in order to comply with specifications in line C) must be 1230/0.61 or 2020 fpm. The following rule and corollary will apply:

Rule—When the damper settings of all the diffusers supplied by a common fan remain constant, any variation in fan delivery will not alter the *percentage* of the total air discharged by each respective diffuser.

Corollary—Any change in the damper settings will alter the air discharge percentages of the individual diffusers.

After dampering and total air delivery adjustments have been completed, a spot check of several random diffusers, together with the base diffuser, using Equations 1 or 2 depending on the type of damper used, will suffice to verify the adequacy of balancing. A traverse by Pitot tube or other convenient device

TABLE 2. BALANCING OF AIR DELIVERY FOR *Example 2*

A Diffuser Location	1	2	3	4	5
D Velocity Reading with All dampers in Full Open Position, fpm	2200	2125	2050	1975	1900
H *Base* Diffuser = Least Value in line D					1900
K Percent of Full Open Damper Capacity Required = $\frac{\text{line H} \times 100}{\text{line D}}$	86.4	89.4	92.7	96.2	100

may be used to verify total air delivery, if desired. For all practical purposes, a tolerance of ±5 percent, either in individual diffuser or total air delivery may be considered satisfactory since the element of human error in the taking of readings may readily account for a difference of this magnitude.

Example 2. An air delivery system shown in Fig. 2 has 5 outlets, each of which should deliver 600 cfm. Illustrate procedure for *balancing* the system.

Solution: In *Example 2* it will be noted that the diffusers are all of the same size and are specified to deliver like capacities. Here the balancing computations may be simplified by dispensing with capacity conversion and using outlet velocities directly as indicated in the Table 2.

Assuming capacity factor for the 10 in. diffusers used in the *Example 2* is 0.31, the correct velocity reading of the *base* diffuser No. 5 after balancing should be 600/0.31 or 1935 fpm.

Other similar variations in calculating procedure will be apparent. For instance, in the case of *Example 1,* while the outlets are of different sizes, the neck velocity values bring all sizes into common relation and these values therefore can be used in a simplified procedure. In any event, the forms suggested in Examples 1 and 2 are applicable to all types of systems and would, therefore, lend themselves to standardization.

Example 3. An air delivery system shown in Fig. 3 has 3 branches with 5 outlets on each branch. The required air volume at each outlet is 600 cfm. Outline the procedure for *balancing* the system.

Solution: The system shown in *Example 3* consists of three branches, and, again entirely in the interest of simplicity, it is assumed that each branch is identical with the single branch used in *Example 2* in number, size and capacity of diffusers. It is further assumed, as may prevail in practice, that the proportioning of trunk and branch ducts alone, without the aid of duct dampers D_a and D_b will not produce uniform apportionment of the total air delivery among the three branches without excessive throttling of the most favored diffusers. In balancing a system of this type by the method proposed in this paper, two alternative procedures can be employed. The diffusers in each branch can be balanced separately as in *Example 2* and the *base* (open damper) outlets will be A_5, B_5 and C_5 respectively. The velocity

TABLE 3. BALANCING OF BRANCH AIR DELIVERY FOR *Example 3*

A Base Diffuser Location	A_5	B_5	C_5
D Assumed Velocity Reading	1900	1780	1560
H Base Branch			1560
Duct Damper	D_b	D_a	
Percentage of Damper Opening = $\frac{\text{line H} \times 100}{\text{line D}}$	82	87.7	

readings of these *base* diffusers after balancing will, however, vary due to the pressure loss imposed by the trunk duct friction between the branches. The difference in capacity between the three *base* diffusers will likewise correspond to the ratio of difference between the air supply to the respective branches. To correct this difference, it is necessary to balance these *base* outlets with respect to each other by the adjustment of duct dampers D_a and D_b. As these dampers are not calibrated, this final balancing operation can only be accomplished by trial and error. However, the required degree of approximate damper adjustment can be computed percentage-wise as given in Table 3, and, with this known, a minimum amount of adjustment will be required.

The system will be balanced when the outlet velocities of the three *base* diffusers are equivalent. When this has been achieved, any of these diffusers can be read for conformity with the specified air capacity and the fan speed adjusted, if necessary, as already described.

The other, and, perhaps more convenient approach to the balancing of *Example 3,* is to first regulate the duct dampers until reasonably uniform outlet velocities are obtained from any three comparable diffusers (one in each branch) and then with these duct damper settings fixed so that they cannot change, to balance all fifteen diffusers as a common system in the manner described for *Example 2.* It will be recognized that the basic method for balancing described is equally applicable to intricate systems involving unequal branches and a multiplicity of diffuser sizes and capacities, requiring only extra steps in computation. Seemingly, systems which are symmetrical and have uniform size diffusers might be balanced on the basis of static pressures within the duct at the diffusers. This, however, is seldom practicable because of the influence of erratic flow patterns.

As already indicated, the authors are of the opinion that special emphasis regarding proper duct design, particularly for air diffusers, is warranted. The original design of a duct distributing system should contemplate the ultimate problem of balancing. The permissible pressure in the duct at the most throttled outlet is, for example, often governed by noise limitation. For extreme quietness this pressure should not exceed 0.25 in. water column. Where noise level is not a factor it may be as high as 0.75 in. water column. Even where noise level is not significant, economy in operation dictates the lowest duct pressure practicable or consistent with duct cost. The design relation between duct pressure and degree of outlet throttling can be expressed by the following Equations 3 and 4:

$$T = 100 - 100\sqrt{\frac{P_1}{P_2}} \quad \ldots\ldots\ldots\ldots \quad (3)$$

$$P_2 = \frac{P_1}{\left(1 - \frac{T}{100}\right)^2} \quad \ldots\ldots\ldots\ldots \quad (4)$$

where

P_1 = minimum duct pressure in inches of water.
P_2 = maximum duct pressure in inches of water.
T = maximum throttling of individual outlet, percent.

Let

$$P_1 = 0.25 \text{ in. WG and } P_2 = 0.75 \text{ in. WG.}$$

Then

$$T = 100 - 100\sqrt{\frac{0.25}{0.75}} = 100 - 57.7 = 42.3 \text{ percent.}$$

Duct Loss = 0.75 − 0.25 = 0.5 in. WG or the equivalent of 500 ft of straight duct based on a loss of 0.1 in. per 100 ft (static regain ignored).

From the preceding, it is obvious that the maximum practical extent of throttling, even disregarding noise, will rarely exceed 40 percent. Whenever it is found impracticable to limit the differential pressure across the outlet to that desired, the system should be sectionalized with duct or splitter dampers (see *Example 3*). In so doing, the throttling in the high pressure portion of the duct is accomplished in two stages and, as a result, the degree of throttling and consequent pressure differential at the outlets is minimized.

Factors which contribute to pressure loss in duct design have long been evaluated and made available for predicting results. However, the use of

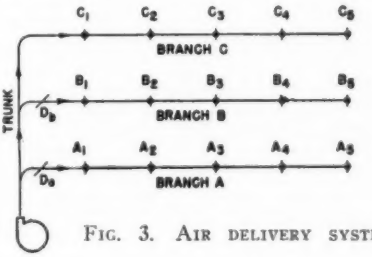

FIG. 3. AIR DELIVERY SYSTEM FOR EXAMPLE 3

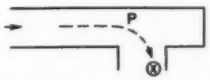

FIG. 4. LOCATION OF HIGH VELOCITY POINT IN DUCT CONNECTION TO DIFFUSER

circular ceiling air diffusers imposes an additional, important, and too often ignored, factor, namely, the behavior of an air stream entering a circular opening in the bottom of a horizontal rectangular duct. Unless the horizontal air velocity gradient across the duct is symmetrical, the air entering the neck of the diffuser will rotate or swirl rapidly, often creating a vortex and, as a consequence, the discharge volume at the prevailing static pressure will be greatly reduced. To regain the specified capacity under such a condition, the duct static pressure would have to be increased correspondingly. When, therefore, the duct design is such that the air does not approach the outlet uniformly, it is impossible to predict the duct pressure necessary to obtain the specified capacity, since the extent and effect of the swirl are unpredictable. It will further be obvious that, in final balancing, a swirl condition at only a single outlet in the system will necessitate an increase in pressure throughout the entire system and may result in increasing the required differential pressure across the upstream outlets prohibitively.

Air directive devices or so-called air equalizing grids placed in the neck or collar of the diffuser at its junction with the bottom of the duct have little effect upon the vortex phenomena and their resultant disadvantages, especially where the outlet is at the end of the duct or branch (and this applies whether the duct is provided with an extension or so-called *cushion head* or not). It

TABLE 4—BALANCING DATA FOR WOOLEN MILL AIR DELIVERY SYSTEM[a]

A	Diffuser Location	A-1	2	3	4	5	6	7	8
D	Velocity Reading	1010	1080	1140	1175	1190	1220	1210	1170
E	Capacity Factor	0.47	0.47	0.47	0.47	0.47	0.47	0.47	0.47
F	Actual Air Delivery (cfm) = line D × line E	475	508	536	552	559	573	569	550
G	Variation = line F/cfm specified	86.4	92.5	97.5	100	102	104	103	100
H	Base (Least value line G)	86.4							
K	Percent Actual Delivery = line H × 100/line G	100	93.5	88.5	86	85	82.5	83.5	86.5
L	Recheck (New line D value)	1150							
M	Corrected Capacity = line E × line L × K/100	541							
N	Error (percent)	−1.3							
A	Diffuser Location	B-1	2	3	4	5	6	7	8
D	Velocity Reading	1150	1150	1255	1260	1350	1350	1350	1280
E	Capacity Factor[b]								
F	Actual Air Delivery (cfm) = line D × line E	540	540	590	592	634	634	634	602
G	Variation = line F/cfm specified	98.2	98.2	107	108	115	115	115	110
H	Base (Least value line G) = Diffuser A-1								
K	Percent Actual Delivery = line H × 100/line G	88	88	80.5	80	75	75	75	79
L	Recheck (New line D Values)			1475					
M	Corrected Capacity = line E × line L × K/100			558					
N	Error (percent)			+2					
A	Diffuser Location	C-1	2	3	4	5	6	7	8
D	Velocity Reading	1155	1270	1275	1275	1310	1355	1380	1210
E	Capacity Factor[b]								
F	Actual Air Delivery (cfm) = line D × line E	543	597	599	599	616	637	649	569
G	Variation = line F/cfm specified	98.6	108	109	109	112	116	118	103
H	Base (Least value line G) = Diffuser A-1								
K	Percent Actual Delivery = line H × 100/line G	87.5	79.5	79	79	77	74.5	73	83.5
L	Recheck (New line D Values)		1460						
M	Corrected Capacity = line E × line L × K/100		545						
N	Error (percent)		−0.5						
A	Diffuser Location	D-1	2	3	4	5	6	7	8
D	Velocity Reading	1260	1350	1375	1425	1445	1470	1480	1270
E	Capacity Factor[b]						0.47	0.47	0.47
F	Actual Air Delivery (cfm) = line D × line E	592	634	646	670	679	691	696	597
G	Variation = line F/cfm specified	108	115	118	122	113	126	127	108
H	Base (Least value line G) = Diffuser A-1								
K	Percent Actual Delivery = line H × 100/line G	80	75	73.5	71	70	68.5	68	79.5
L	Recheck (New line D Values)				1600				
M	Corrected Capacity = line E × line L × K/100				535				
N	Error (percent)				−2.4				

TABLE 4 (CONTINUED)—BALANCING DATA FOR WOOLEN MILL AIR DELIVERY SYSTEM[a]

TABLE 4 (CONTINUED)—BALANCING DATA FOR WOOLEN MILL AIR DELIVERY SYSTEM[a]

A	Diffuser Location	E-1	2	3	4	5	6	7	8
D	Velocity Reading	1285	1385	1440	1535	1535	1550	1525	1390
E	Capacity Factor[b]							0.47	0.47
F	Actual Air Delivery (cfm) = line D × line E	604	651	677	721	721	728	717	653
G	Variation = line F/cfm specified	110	118	123	131	131	132	130	119
H	Base (Least value line G) = Diffuser A-1								
K	Percent Actual Delivery = line H × 100/line G	78.5	73	70	66	66	65	66	72.5
L	Recheck (New line D Values)							1755	
M	Corrected Capacity = line E × line L × K/100							544	
N	Error (percent)							−0.8	
A	Diffuser Location	F-1	2	3	4	5	6	7	8
D	Velocity Reading	1255	1380	1390	1400	1480	1460	1495	1385
E	Capacity Factor[b]							0.47	0.47
F	Actual Air Delivery (cfm) = line D × line E	590	649	653	658	696	686	703	651
G	Variation = line F/cfm specified	107	118	119	120	127	125	128	118
H	Base (Least value line G) = Diffuser A-1								
K	Percent Actual Delivery = line H × 100/line G	80.5	73	72.5	72	68	69	67.5	73
L	Recheck (New line D Values)								1610
M	Corrected Capacity = line E × line L × K/100								552
N	Error (percent)								+0.8
A	Diffuser Location	G-1	2	3	4	5	6	7	8
D	Velocity Reading	1225	1240	1340	1395	1400	1430	1455	1400
E	Capacity Factor[b]							0.47	0.47
F	Actual Air Delivery (cfm) = line D × line E	576	583	630	657	658	672	684	658
G	Variation = line F/cfm specified	105	106	114	119	120	122	124	120
H	Base (Least value line G) = Diffuser A-1								
K	Percent Actual Delivery = line H × 100/line G	82.5	81.5	75.5	72.5	72	70.5	69.5	72
L	Recheck (New line D Values)		1490						
M	Corrected Capacity = line E × line L × K/100		570						
N	Error (percent)		+4						
A	Diffuser Location	H-1	2	3	4	5	6	7	8
D	Velocity Reading	1160	1200	1295	1300	1340	1325	1375	1410
E	Capacity Factor[b]							0.47	0.47
F	Actual Air Delivery (cfm) = line D × line E	545	564	609	611	630	623	646	663
G	Variation = line F/cfm specified	99.1	102	111	111	114	113	118	120
H	Base (Least value line G) = Diffuser A-1								
K	Percent Actual Delivery = line H × 100/line G	87	84	78	77.5	75.5	76	73.5	71.5
L	Recheck (New line D Values)	1350							1580
M	Corrected Capacity = line E × line L × K/100	552							532
N	Error (percent)	+0.8							−3

[a] All diffusers were of 12 in. size (neck diameter). Each outlet was required to deliver 550 cfm. Line F=Total=39869 cfm, average=623 cfm, error = +13 percent. Line M=Total (8 outlets) 4929 cfm, average=547.7, error=(−0.46) percent. [b] As all diffusers are of same size, the capacity factor is constant (0.47).

is the authors' experience that the only satisfactory and reliable preventive for air entry turbulence is the use of properly designed air directing and straightening vanes in all elbows and branch take-offs to insure symmetrical approach velocity to all diffusers. This caution applies likewise to *fixed throat take-offs* because the design of such take-offs is usually predicated upon the hypothesis of symmetrical air flow across the approach duct. The effect of erratic air patterns produced by take-offs will be noted in the illustration of the balancing of an actual system in the section Application of Method.

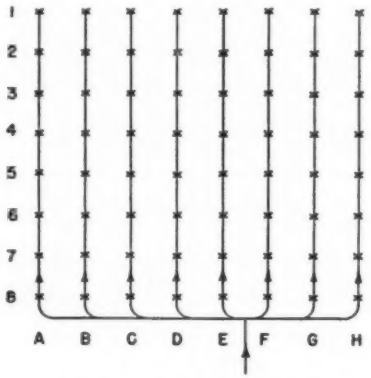

FIG. 5. SCHEMATIC DIAGRAM AIR DISTRIBUTING SYSTEM IN A WOOLEN MILL

Another factor affecting the total pressure calculations which enters into diffuser system duct design is the static pressure required by an outlet collar independent of the resistance of the diffuser itself (see Fig. 4).

Due to the change in air direction, a high velocity strata at point x is unavoidable and its effect upon duct static pressure may be expressed by Equation 5.

$$P = 2.2\left(\frac{V}{4005}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots \quad (5)$$

where

P = duct static pressure in inches of water.

V = *mean* neck velocity in feet per minute or $\frac{\text{cfm}}{\text{Neck Area}}$

The value 2.2 remains practically constant for all outlets in a conventional duct system as the ratio of duct static pressure to duct velocity pressure at each outlet will be approximately equal. Unless the diffuser manufacturer's resistance ratings include this factor, it must be allowed for in the calculation of total static pressure.

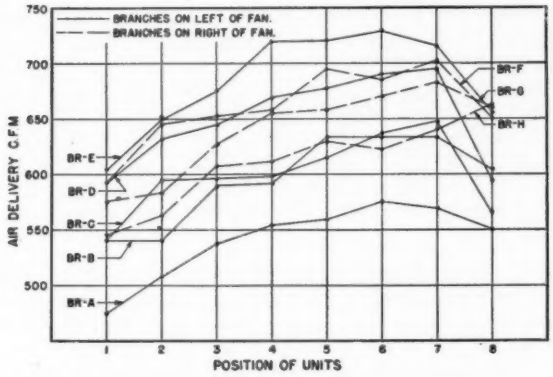

FIG. 6. AIR DELIVERY OF EACH DIFFUSER IN FIG. 5 BEFORE BALANCING

APPLICATION OF METHOD

The following describes the actual procedure in balancing a system in a woolen mill. This installation included 64 diffusers each specified to deliver 550 cfm, and distributed as shown in schematic diagram, Fig. 5. (It should be noted that the only dampers used in balancing this system are those in the individual diffusers.)

Table 4 is the actual tabulation of the balancing data arranged in the form already explained. From this it will be seen that the original maximum extent of unbalance was 35 percent.

Table 5 is a tabulation of the diffusers by branches with the total volume (cfm) of each branch entered in the right-hand column and the total volume (cfm) for the series of diffusers occupying the same relative position in the

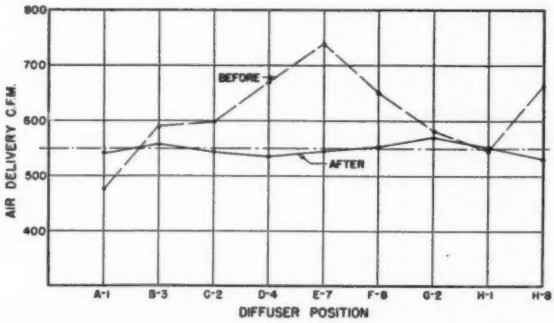

FIG. 7. AIR DELIVERY OF NINE ARBITRARILY SELECTED DIFFUSERS FROM FIG. 5—AFTER ADJUSTMENT

TABLE 5. DELIVERY VOLUME BY BRANCHES IN CUBIC FEET PER MINUTE

(For System shown in Fig. 5)

Diffuser Position	1	2	3	4	5	6	7	8	Totals
Branch A.....	475	508	536	552	559	573	569	550	4322
Branch B.....	540	540	590	592	634	634	634	602	4766
Branch C.....	543	597	599	599	616	637	649	569	4809
Branch D.....	592	634	646	670	679	691	696	597	5205
Branch E.....	604	651	677	721	721	728	717	653	5472
Branch F.....	590	649	653	658	696	686	703	651	5286
Branch G.....	576	583	630	657	658	672	684	658	5118
Branch H.....	545	564	609	611	630	623	646	663	4891
Totals.....	4465	4726	4940	5060	5193	5244	5298	4943	39869

branches entered at the bottom. These data indicate the effect of duct friction loss. As would be expected, the air delivery through the branches farthest from the fan is less than through branches nearer the fan. Likewise, the diffusers at the far end of the branches deliver less air than those at the near end. The exceptions in this instance are those diffusers close to the branch take-offs (except unaccountably in Branch H) which, it will be noted, deliver less air than those next downstream. This variation is due to the swirl imparted to the air by the elbows, an effect already pointed out in preceding text. Thus, the aggregate volume (cfm) of the diffusers in position 8 (Fig. 5) which are located immediately beyond the take-offs is 4943 cfm while the aggregate volume of the outlets in position 7 where the air flow has smoothed out, is 5298. From position 6 to the end of the ducts, the air delivery decreases consistently as affected by duct friction loss. The total volume (cfm) delivered by the system before balancing was 39,869 or 13.3 percent higher than specified.

Fig. 6 shows the volume (cfm) delivered by each diffuser in any given branch and is simply a visual representation of the tabulation, Table 5.

After damper adjustments were completed, check readings of approximately one in every eight diffusers were made to confirm that the system was balanced. Reference to Table 4 will reveal that, after balancing, the maximum unbalance (extreme difference in air delivery between any two individual diffusers) was 7 percent and well within the tolerance of $\pm$ 5 percent considered acceptable. Also, the average air delivery of the nine diffusers rechecked was 547.6 cfm per diffuser, indicating that the system as a whole was approximately 0.5 percent under capacity, a discrepancy too slight to warrant adjustment of the fan speed.

Fig. 7 is a plot of the delivery characteristic of the nine diffusers rechecked, before and after balancing, and demonstrates the degree of balance achieved directly in a single adjustment.

Referring to the simplicity of the balancing method described in this paper, it may be mentioned that the balancing of this system, including desk and ladder operations, required but 24 man hours.

In conclusion, the authors wish to stress that the data presented in this paper are intended to be introductory rather than conclusive and particularly that investigation of the possible extent of error inherent in the method described is suggested as a suitable project for A.S.H.V.E. research.

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No. 1362

AIR DISTRIBUTION AND DRAFT

By JOHN RYDBERG* AND PER NORBÄCK**, STOCKHOLM, SWEDEN

AMONG the numerous ways of introducing ventilation air into rooms, there are only a few cases in which the air jet course can be determined mathematically. Nevertheless, it has been found that a whole group of discharge devices, for practical purposes, may be satisfactorily computed in advance, both as regards the course of jet and the risk of draft; these are the cases where the air is discharged through concentrated outlets, such as circular or rectangular holes or grids of various designs. The detailed shape of the hole or grid need not be known. The only condition is that the jet is not disturbed by surrounding walls, that the ratio between the longest and shortest side is not too great and that no guide vanes are employed to give the air a special direction.

SIMILARITY CONSIDERATIONS

The discharged air is assumed to have the same temperature as the room air. For free air jets from similar outlets, the shape of the jets will be similar if the Reynolds numbers N_{Re} are the same. This means that the velocity V in equally located points will be proportional to the outlet velocities V_o. Two points in different jets are equally located if x/d, y/d, and z/d are equal for both systems (see Fig. 1).

If then, N_{Re}, x/d, y/d, and z/d are equal for different systems, then V/V_o will also have the same value in these systems. Generally, therefore, if f expresses a general function, it is possible to write Equation 1 for jets from similar discharge outlets

$$\frac{V}{V_o} = f\left(N_{Re}, \frac{x}{d}, \frac{y}{d}, \frac{z}{d}\right) \quad \ldots\ldots\ldots\ldots \quad (1)$$

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A jet from a circular opening will be symmetric about an axis, and, in this case, one of the cobrdinates drops out, and then Equation 1 becomes

$$\frac{V}{V_0} = f\left(N_{Re}, \frac{x}{d}, \frac{y}{d}\right) \quad \ldots\ldots\ldots\ldots \quad (2)$$

It is found in practice that the effect of N_{Re} is so insignificant that it can be ignored. This can be shown, if it is assumed that the pressure in the jet is constant. This assumption is fulfilled with small error for a free jet, as has been confirmed by measurements. If the pressure is constant then, according to the principle of momentum conservation, in each cross section of the jet,

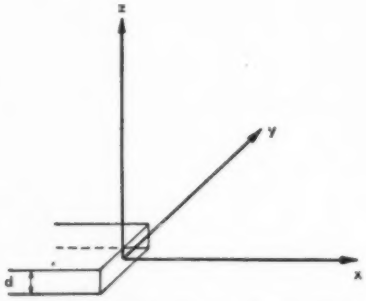

FIG. 1. DIAGRAM FOR LOCATING POINTS IN A JET OF AIR

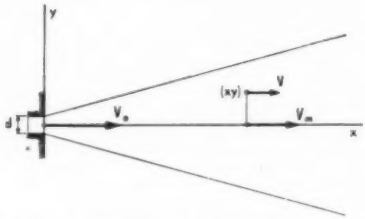

FIG. 2. RELATION OF PRODUCT OF VELOCITY AND MASS IN DIFFERENT POINTS IN AIR STREAM

the product of the velocity and the mass flowing forward per unit of time will be constant. Equation 3 then applies for a circular jet. See Fig. 2.

$$V_0^2 \frac{\pi d^2}{4} = \int_0^\infty V^2 2\pi y\, dy \quad \ldots\ldots\ldots\ldots \quad (3)$$

or,

$$\int_0^\infty \left(\frac{V}{V_0}\right)^2 \frac{y}{d}\, d\left(\frac{y}{d}\right) = 1/8 \quad \ldots\ldots\ldots\ldots \quad (4)$$

Equation 4 can only be satisfied by an expression in the form

$$\frac{V}{V_0} = f\left(\frac{y}{d}\right) \quad \ldots\ldots\ldots\ldots \quad (5)$$

The assumption that the pressure is constant means that, for a given section (*i.e.*, for a given value of x/d), V/V_0 is only dependent on y/d, and, consequently, independent of N_{Re}. Therefore

$$\frac{V}{V_0} = f\left(\frac{x}{d}, \frac{y}{d}\right) \quad \ldots\ldots\ldots\ldots \quad (6)$$

and also

$$\frac{V}{V_0} = f\left(\frac{x}{d}, \frac{y}{x}\right) \quad \ldots\ldots\ldots\ldots \quad (7)$$

In the center of the jet $y/x = 0$, so that

$$\frac{V_m}{V_o} = f\left(\frac{x}{d}\right) \quad \ldots \ldots \ldots \ldots \ldots \quad (8)$$

For a given cross section of the jet, *i.e.*, for a given value of x/d, Equation 9 applies

$$\frac{V}{V_o} = f\left(\frac{y}{x}\right) \quad \ldots \ldots \ldots \ldots \ldots \quad (9)$$

Or, when $x/d =$ constant, $V_m/V_o =$ constant,

$$\frac{V}{V_m} = f\left(\frac{y}{x}\right) \quad \ldots \ldots \ldots \ldots \ldots \quad (10)$$

The application of Equation 10 is limited to a given value for x/d.

In practice it is found that for large values of x/d, *i.e.*, for such great distance from the outlet that this may be regarded as a point, the last named function is independent of the value for x/d.[1, 2, 3, 4]

For large values of x/d (great distance from the opening or small openings), it is therefore possible to write

$$\frac{V}{V_o} = \phi\left(\frac{y}{x}\right) \Psi\left(\frac{x}{d}\right) \quad \ldots \ldots \ldots \ldots \quad (11)$$

where ϕ and Ψ denote definite functions.

Equation 12 is the result of substituting the expression for V/V_o in Equation 4, derived previously

$$\frac{x^2}{x^2} \Psi^2\left(\frac{x}{d}\right) \int_o^\infty \phi^2\left(\frac{y}{x}\right) \frac{y}{x}\, d\left(\frac{y}{x}\right) = 1/8 \quad \ldots \ldots \quad (12)$$

As the integral comprised in Equation 12 is constant,

$$\Psi\left(\frac{x}{d}\right) = K_1 \frac{d}{x} \quad \ldots \ldots \ldots \ldots \quad (13)$$

and thus, after substitution in Equation 11,

$$\frac{V}{V_o} = K_1 \frac{d}{x} \phi\left(\frac{y}{x}\right) \quad \ldots \ldots \ldots \ldots \quad (14)$$

or

$$\frac{V_m}{V_o} = K_1 \frac{d}{x} \quad \ldots \ldots \ldots \ldots \quad (15)$$

$$\frac{V}{V_m} = \phi\left(\frac{y}{x}\right) \quad \ldots \ldots \ldots \ldots \quad (16)$$

It is advisable to determine empirically the constant K_1 in Equation 15 and the function ϕ must also be determined in the same manner. The constant in question may, in accordance with experimental values of Zimm[2], Trüpel[3], Cleve[4], Kuethe[5], and Ruden[6], be taken for the area nearest the outlet at 6 to 7. McElroy[7] has arrived at the mean value of 5.3. Tuve[8] found that for the first part of the jet, where velocity exceeded 500 fpm, the value of K_1 was approximately 5.5. At the end of the jet, however, where the velocity lay below

500 fpm, it was found correct to use lower values for the constant. This decrease of K_1, as stated by both McElroy and Tuve, possibly has its cause in disturbances from unconsidered air currents in the rooms where the measurements were made. For a velocity of jet of 200 fpm the value of the constant, according to Tuve, will be 4.5 and for a velocity of 100 fpm it will be 3.5. Rydberg and Norbäck[9] obtained a mean value of about 4.3.

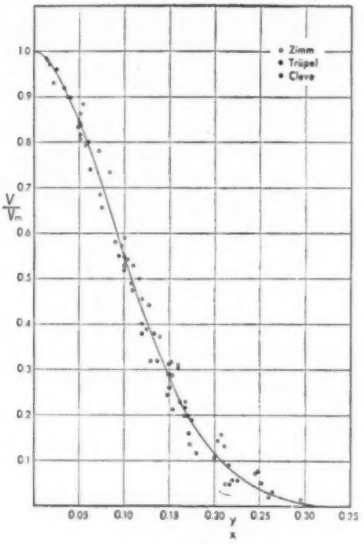

FIG. 3. VALUE OF $\frac{V}{V_m}$ AS FUNCTION OF $\frac{y}{x}$

In computations for air conditioning systems, it should be possible, as a reasonable mean value, to suggest the value 5 for the constant K_1 and then

$$\frac{V_m}{V_o} = 5\,\frac{d}{x} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad (17)$$

if V_m and V_o are measured in the same units, and the same applies for d and x respectively.

The function given in Equation 16 is shown in Fig. 3 with experimental values taken from Zimm[2], Trüpel[3], and Cleve[4].

It will be seen from the foregoing that the angle of a circular jet is constant and amounts to approximately 35 deg.

The mean velocity in any cross section of the jet is about 0.2 V_m.

AIR DISCHARGED THROUGH GRID

Fig. 4 shows in diagram a jet issuing through a grid. Model experiments made by Rydberg and Norbäck[9] have shown that immediately after the separate jets leave the grid a contraction takes place and the separate jets attain their greatest velocity in the section k—k, Fig. 4, from which the velocity is diminished through entrainment of surrounding air, and the separate jets unite quickly into one homogeneous jet. At a comparatively small distance from the grid, in the section b—b, Fig. 4, the velocity becomes uniform and almost constant throughout the whole cross section of the combined jet. Even if the grid is rectangular and the ratio between the longest and shortest sides is not too great, the jet in the cross section b—b, Fig. 4, has assumed a practically

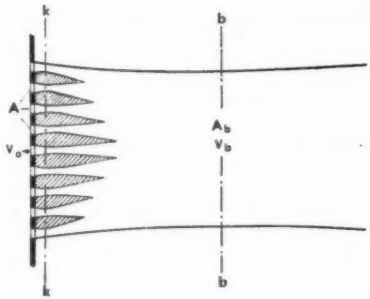

FIG. 4. SHAPE OF JETS LEAVING A GRID

circular cross section. This was demonstrated by Tuve[8] and others. The section b--b lies at a distance x_b from the grid that is of the same order of magnitude as the longest dimension of the grid.

The following symbols are used

A = the free area of the grid (the total area of openings).
V_o = the velocity in the grid openings.
A_k = the total area of the separate jets in section k — k, Fig. 4.
V_k = the velocity of the separate jets in section k — k, Fig. 4.
A_b = area of jet in section b — b, Fig. 4.
d_b = diameter of jet in section b — b, Fig. 4.
V_b = velocity of jet in section b — b, Fig. 4.
V_m = velocity in center at distance x from the grid.
α = coefficient of discharge for the grid.

In the section b—b, Fig. 4, the jet resembles a homogeneous jet that has just left a circular nozzle and the formulas for circular jets should apply to the jet in question beyond section b—b. Then (as per Equation 17)

$$V_m = 5 \frac{V_b d_b}{x - x_b} \quad \ldots\ldots\ldots\ldots \quad (18)$$

or, since

$$A_b = \frac{\pi d_b^2}{4} \quad \ldots \quad (19)$$

$$V_m = 5 \frac{V_b}{x - x_b} \sqrt{\frac{4 A_b}{\pi}} \quad \ldots \quad (20)$$

In the separate jets, the pressure drops from the grid to the section k—k. Beyond the section k—k there first prevails a certain vacuum between the separate jets, but the pressure drop between sections k—k and b—b is insignificant and it is therefore possible to apply in this area the principle of momentum, which gives

$$V_k^2 A_k = V_b^2 A_b \quad \ldots \quad (21)$$

Since x_b is usually small in comparison with x, the result, after eliminating V_b and A_b is

$$V_m = 5 \frac{V_k}{x} \sqrt{\frac{4 A_k}{\pi}} \quad \ldots \quad (22)$$

If the coefficient of discharge for the grid openings is α, Equations 23 and 24 may be written

$$A_k = \alpha A \quad \ldots \quad (23)$$

$$V_o = \alpha V_k \quad \ldots \quad (24)$$

and therefore

$$V_m = 5 \frac{V_o}{x} \sqrt{\frac{4 A}{\pi \alpha}} \quad \ldots \quad (25)$$

For a circular opening in a thin wall, α is approximately 0.60. For a square opening in a thin wall, according to Tuve[8], $\alpha = 0.61$, and the same figure applies also for rectangular openings, if the ratio between the long and the short side is not too great. When that ratio grows, α increases, and for openings where the long side is 12 or 48 times the short side, α becomes, according to the same source, 0.64 and 0.67 respectively. For grids, α assumes still greater values. Tuve gives for different types of commercial grids α values from 0.72 and 0.87 with a mean of 0.8. For the grids investigated by Rydberg and Norbäck α was found to be 0.64. That the value for this type of grid was relatively low would seem to be due to grid construction employing circular, sparsely situated holes.

As a mean value for rough estimates, a value of $\alpha = 0.7$ seems reasonable. With this value,

$$V_m = 6.7 \frac{V_o \sqrt{A}}{x} \quad \ldots \quad (26)$$

If air volume per unit of time $Q = V_o A$, is substituted for air velocity, Equation 27 is obtained

$$V_m = 6.7 \frac{Q}{x \sqrt{A}} \quad \ldots \quad (27)$$

The constant 6.7 is dimensionless.

The two last formulas apply for free jets from concentrated discharge openings of rather indefinite form, *e.g.* circular or rectangular holes, grids of various constructions, etc. In grids, the form of the holes or the distance between them does not matter. For grids made with guide vanes which give the air a special direction or have a diffusing effect, however, the formulas may not apply.

The same formulas apply for single openings and for grids because the decrease in velocity due to dilution immediately upon leaving the grid is balanced by an increase in total jet area.

TEMPERATURE DISTRIBUTION IN AIR JETS

For an air particle with velocity V in any point of the jet shown in Fig. 5, part of its mass is obtained from the air (of mass m) discharged with velocity V_o through the opening and the remainder is obtained from the still air (of mass

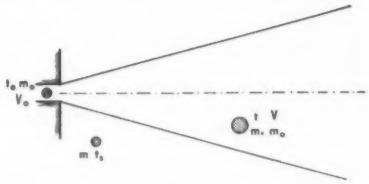

FIG. 5. TEMPERATURE AND MOMENTUM OF A PARTICLE IN AN AIR JET

m_o) surrounding the jet. If the pressure in the jet is constant and friction is ignored, the momentum of the arbitrarily selected air particle with the mass m + m_o must be equal to the momentum of the particle m_o

$$m_o V_o = (m + m_o)\, V \quad \ldots\ldots\ldots\ldots \quad (28)$$

If the temperature of the discharged air is t_o, the temperature of the room is t_s, and if the temperature in the arbitrarily selected point in the jet is t, there is obtained, if heat exchange by conduction is ignored,

$$m_o t_o + m t_s = (m + m_o) t \quad \ldots\ldots\ldots\ldots \quad (29)$$

If, in Equations 28 and 29, m and m_o are eliminated, Equation 30 is obtained

$$\frac{t - t_s}{V} = \frac{t_o - t_s}{V_o} \quad \ldots\ldots\ldots\ldots \quad (30)$$

Equation 30 states that the ratio of the excess temperature (the difference between the temperature in the jet and the surrounding air) and the velocity is constant throughout the jet and equal to the same ratio computed for the outlet. This has also been confirmed experimentally[6].

By combining Equations 26 and 30

$$t_m - t_s = 6.7\,(t_o - t_s)\,\frac{\sqrt{A}}{x} \quad \ldots\ldots\ldots\ldots \quad (31)$$

where t_m = the temperature in the center of the jet at distance x from the outlet.

THE EFFECTIVE TEMPERATURE IN AN AIR JET

The effective temperature t_e for a given case with arbitrary values for the air temperature t, the wall temperature t_v, the dew point temperature t_d and the air velocity V, is defined as the temperature required in a space in order to have the same heat loss from the human body as in the case given, provided that, in the space considered, the air and wall temperature are equal, that the relative air humidity is 100 percent ($t_d = t$) and that the air velocity is equal to zero.

In a small temperature range, it is possible to count on the effective temperature as varying rectilinearly with the air temperature, the wall temperature, the dew point temperature and the air velocity. It is then possible to write for the effective temperature in any point of the jet within a small temperature range,

$$t_e = At + Bt_v + Ct_d - DV \quad \ldots\ldots\ldots\ldots \quad (32)$$

A, B, C and D are constants and $A + B + C = 1$. If the effective temperature for the still air outside the jet is designated t'_e and the dew point temperature for the same air is designated t'_d there is obtained

$$t'_e = At_s + Bt_v + Ct'_d \quad \ldots\ldots\ldots\ldots \quad (33)$$

The difference between the effective temperatures of the surrounding air and the jet, the effective temperature drop of the jet, will be

$$t'_e - t_e = A(t_s - t) + C(t'_d - t_d) + DV \quad \ldots\ldots\ldots \quad (34)$$

Dividing Equation 34 by A

$$\theta = \frac{t'_e - t_e}{A} = (t_s - t) + \frac{C}{A}(t'_d - t_d) + \frac{D}{A}V. \quad \ldots\ldots \quad (35)$$

The quantity θ obtained in this manner is proportional to the effective temperature drop of the jet and differs from this only in that its scale is so selected that it coincides with the scale for the air temperature t.

The quantity θ indicates, in the first place, the direct difference between the real air temperature of the surroundings and of the jet. In addition, the air temperature difference, which, in respect to cooling, is equivalent to the difference in dew point temperature between the air of the room and the air of the jet, and finally the air temperature difference which concerns cooling effect, is equivalent to the air velocity in the jet.

To sum up, it may be said that θ indicates the difference between the room air temperature and the temperature of still room air which would be required to give the same cooling effect on the human body as the air jet. Thus, θ represents a kind of effective air temperature drop.

Usually $(t'_d - t_d)$ is small. Only if large quantities of moisture in proportion to the volume of air are delivered to, or extracted from the air in the room, will the difference be of significance. Consequently, in most cases, the effect of humidity may be ignored, and,

$$\theta = t_s - t + \frac{D}{A}V \quad \ldots\ldots\ldots\ldots \quad (36)$$

The constant $\frac{D}{A}$ in this equation may be determined empirically.

If $\theta = 0$,

then,

$$\frac{t - t_s}{V} = \frac{D}{A} \quad \ldots\ldots\ldots\ldots \quad (37)$$

According to the foregoing, in an air jet the ratio given in Equation 37 between the excess temperature and the velocity is constant throughout the jet and equal to the same ratio at the outlet. If an air jet is so arranged that in the area where the velocity is low ($V < 200$ fpm) and where, therefore, Equation 32 may be considered applicable, the air temperature drop, θ, will be equal to zero, as may be shown by test. The value of the constant can be determined from data for the temperature and velocity of the discharged air[10].

Investigations of this nature for arriving at the value of the constant D/A have been carried out at the Technical University at Stockholm by Norbäck[11]. For these tests subjects were placed in a horizontal jet of hot air and the velocity and the excess temperature of the jet regulated so that the jet, according to statements by the subjects, caused the same temperature feeling as the surroundings. According to Equation 30, the ratio between excess temperature and velocity is constant in all parts of the jet, and, therefore, it is of no consequence where the subject is placed in the jet, provided it is done in those parts of the jet where the deviations from the surroundings are small and thus Equation 32 is satisfied.

In Fig. 6 there is shown the relation obtained by Norbäck between the excess temperature $t - t_s$ and the air velocity V which gives $\theta = 0$, *i.e.*, freedom from draft. At the same time, there is given in Fig. 6 the corresponding relation computed from experimental values given in tests by Houghten and Yaglou[12, 13, 14, 15, 16] and Weiss[17].

The proper relation, according to Fig. 6, should probably be represented by a slight curve, but, for low velocities, a rectilinear relation in the manner given earlier is satisfactory. The constant D/A will be approximately equal to 8, when the temperature is measured in Centigrade degrees and the velocity in meters per second (if the temperature is measured in Fahrenheit degrees and the velocity in feet per minute, the value of the constant will be 0.07).

The effective air temperature drop of the jet, in general, when the temperature is measured in Fahrenheit degrees and the velocity in feet per minute, is given by the following formula.

$$\theta = t_s - t + 0.07\,V \quad \ldots\ldots\ldots\ldots \quad (38)$$

For the center of a jet issuing from a grid or other single opening, if the foregoing is taken into account, there is obtained

$$\theta = 6.7\,(t_s - t_o) + 0.07\,V_o\,\frac{\sqrt{A}}{x} \quad \ldots\ldots\ldots \quad (39)$$

As the greatest air velocities and temperature drops arise in the jet center, Equation 39 is usually sufficient for judging the possibility of draft.

By appropriate selection of the outlet area A, the distance x between the discharge openings and the zones where people are stationed, as well as the discharge air temperature t_o and the velocity V_o, it is obviously possible to attain a value for the effective air temperature drop of the jet, θ, that may be tolerated from the draft point of view. Sensitivity to temperature drops of this kind may vary. Norbäck's experiments suggest that generally an unfavorable reaction will be obtained if the value of θ exceeds 2 to 3 F deg. It is assumed that the temperature of the surrounding walls and the absolute humidity of the

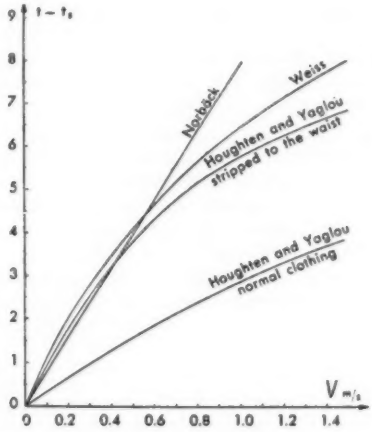

FIG. 6. RELATION BETWEEN AIR VELOCITY AND TEMPERATURE DROP REQUIRED FOR FEELING OF EQUAL WARMTH

air are kept constant. By these last assumptions concerning the radiation conditions and air humidity, the action of the effective temperature drop of the jet on the effective room temperature is diminished in its effect on the total heat loss from the human body.

Equation 39 is only applicable in zones where the air velocities are less than 200 fpm.

On the discharge of heated or cold jets, the gravitational forces give additional velocities which may cause considerable deviations from the isothermal jet course. In the formulas given, no account is taken of this effect. Proper corrections can be made both for horizontal and vertical jets.

FREEDOM FROM DRAFT ON DISCHARGE OF HOT AIR

It is seen from the foregoing that freedom from draft, *i.e.*, $\theta = 0$, is attained if

$$\frac{t - t_s}{V} = 0.07 \quad \ldots\ldots\ldots\ldots\ldots \quad (40)$$

Since the ratio between the excess temperature and the velocity is constant throughout the jet, this condition may be written

$$\frac{t_o - t_s}{V_o} = 0.07 \quad \ldots\ldots\ldots\ldots\ldots (41)$$

where, therefore, t_o *and* V_o are the Fahrenheit temperature and velocity in feet per minute in the discharge opening and t_s is the temperature of the surrounding still air in Fahrenheit degrees.

For hot air discharge, complete freedom from draft can be insured if the foregoing condition is fulfilled. In addition to the condition of Equation 41, there are other conditions that must also be met in practice. Difficulty in this respect, however, is not encountered as a rule. Usually the ventilation air required for an installation is given by

$$Q = A\ V_o\ c_p\ (t_o - t_s) \quad \ldots\ldots\ldots\ldots\ldots (42)$$

where

Q = the amount of heat required.
A = the outlet area.
c_p = the specific heat of the air.

The conditions of Equations 41 and 42 determine both the excess temperature of the discharge air and the velocity, if the area A is fixed. If Q varies, *e.g.,* with the season, then both excess temperature and velocity, *i.e.,* the air volume, must be capable of variation. One way of achieving the stated conditions without varying the air volume is to alter the area A. In most cases, however, this is not possible, and, therefore, the air volume must be varied. If the fresh air volume is fixed, which is another common condition, the discharge air volume can still be varied at will, if circulating return air is mixed with it.

MINIMUM DRAFT FOR COLD AIR DISCHARGE

With discharge of cold air, it is not possible to assure freedom from draft. The cooling action due to the air movement is in this case no longer compensated by an excess temperature in the jet, but, on the contrary, is accentuated, owing to the jet having, throughout its course, a lower temperature than the air of the room.

For the amount of refrigeration introduced by the discharge air into the premises per unit of time, Equation 43 applies

$$W = A\ V_o\ c_p\ (t_s - t_o) \quad \ldots\ldots\ldots\ldots\ldots (43)$$

If A is eliminated between Equations 39 and 43,

$$\theta = \frac{6.7}{\sqrt{c_p}} \cdot \frac{\frac{t_s - t_o}{V_o} + 0.07}{\sqrt{\frac{t_s - t_o}{V_o}}} \cdot \frac{\sqrt{W}}{x} \quad \ldots\ldots\ldots\ldots (44)$$

If, for a constant value of the refrigeration load W, θ is differentiated with respect to the quantity $\frac{t_s - t_o}{V_o}$

$$\frac{d\,\theta}{d\left(\frac{t_s - t_o}{V_o}\right)} = \frac{6.7}{\sqrt{c_p}} \cdot \frac{\frac{t_s - t_o}{V_o} - 0.07}{2\left(\frac{t_s - t_o}{V_o}\right)^{3/2}} \cdot \frac{\sqrt{W}}{x} \quad \ldots\ldots\ldots (45)$$

Inserting the derivative = 0, the condition for minimum value of θ is obtained (temperature in Fahrenheit degrees and velocity in feet per minute)

$$\frac{t_s - t_o}{V_o} = 0.07 \quad \ldots\ldots\ldots\ldots\ldots (46)$$

When the refrigeration load is given and constant, and if the foregoing condition is fulfilled, then θ obtains a minimum value, *i.e.* the feeling of draft for a person who encounters the jet will be the smallest possible.

This can also be expressed so as to show that, for a given admissible value for the feeling of draft, the throw of the jet has a minimum.

As stated, it is only if the refrigeration load is constant that a minimum is obtained. This condition of constant refrigeration load is almost always obtained in practice. The refrigeration load may be allowed to vary, *e.g.*, with the season, and the only requirement is that, at any given time, there shall be a given refrigeration load. As the refrigeration load itself does not enter into the condition for minimum, this condition has the same form for different occasions with different refrigeration loads, even if the values comprised for temperature drops and velocity do vary.

It will be seen from Equation 44 that θ increases with increasing value for the refrigeration load W and that it decreases with increasing x.

To assure that the draft in a given case shall be as small as possible, it is evident that the refrigeration W should be brought down to the lowest possible value (if in this connection the geometric shape of the premises, of which it is not possible to make a general statement, is ignored). After W has been fixed at as low a value as possible, the most favorable result is obtained if the minimum condition given is fulfilled.

The reason why a minimum is obtained is that, if the difference of temperature selected for a certain definite refrigeration load is small, the air volume must be made large, and then the velocity or inlet area will be large and the jet will have a long throw. If, on the other hand, a large temperature difference is taken the jet will be cold and will cause great cooling effect, even if the velocity is small.

If the inlet area A is given, the temperature drop and velocity of the discharge air are determined by Equations 43 and 46. If the refrigeration load W varies from time to time, *e.g.* with the season, the foregoing data for discharge air must be varied accordingly, if a minimum of draft is to be assured.

If there is also a demand that the fresh air volume be adaptable to requirements, *e.g.*, dependent on the occupation of the premises, this demand can be met if the area A be made variable. Usually A cannot be altered and then there remains only the possibility of varying the discharge air volume irrespective of the fresh air volume by mixture with circulating return air. It is therefore possible, by employment of return air, to obtain a minimum of draft also in the general case, when both the refrigeration load and the fresh air

volume are given for each occasion, but can be altered irrespective of each other.

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DISCUSSION

C. M. HUMPHREYS, Cleveland, Ohio (WRITTEN): This paper presents an interesting treatment of the performance of air streams issuing from grills and similar openings. The proposed method of predicting draft is open to question, however, and any attempt to make practical application of the equations given for this purpose should be made with extreme caution.

The Authors limit the application of their equations to jets which are not disturbed by surrounding walls and in which no guide vanes are employed. Although it is not so stated in the paper, normal and uniform approach to the outlet must also be a prerequisite. Such limitations would preclude the use of the proposed method in the actual design of air conditioning systems.

The value of K_1 (Equations 13, 14 and 15) chosen by the authors does not seem to be consistent with other limitations which they impose. Various investigators have found values for K_1 varying from 7 to 3.5. G. L. Tuve obtained values of 5.5, and 4.5 and 3.5 for 500, 200 and 100 fpm, respectively. McElroy also found that K_1 decreased with decreasing velocity. Although the authors state that their equations for draft are applicable only to velocities lower than 200 fpm, they have used a value of K_1 which would be indicated for almost twice that maximum velocity.

Previous work by the A.S.H.V.E. Research Laboratory would indicate a much more complex relationship between drafts, air velocity, and temperature difference than that derived by the authors. It was found, for instance, that for a given difference between room and jet temperature, the air velocity at the ankles could be approximately three times as great as that which could be tolerated at the back of the neck.

The results obtained by Houghten and Yaglou and shown in Fig. 6 would suggest that the type and amount of clothing worn also is of importance in any consideration of drafts.

It seems unlikely that draft is entirely independent of room or jet temperatures, and that, as indicated in the paper, it depends only on the jet velocity and the difference between the room and jet temperatures. According to Equation 41 with jet velocities of 100 fpm, equivalent conditions of draft would be obtained with room and jet air temperatures of either 68 F and 75 F or 75 and 82 F, respectively.

In closing, I should like to congratulate the authors for their courage in undertaking an analysis of drafts, when we in this country have not even been able to agree on the definition of a draft.

H. B. NOTTAGE, Cleveland, Ohio (WRITTEN): Oversimplification of complicated problems, as evidenced in this paper, has long been a practical expedient in air distribution for need of accurate information. This is acceptably justified by urgent necessity, although this paper presumably would not have taken its present assertive form if information such as the following had been known to the authors:

1 The Reynolds Number in Equation 1 has been shown by other experiments to have an appreciable influence.

2 Careful aerodynamics research has given further evidence of the nonconstancy of the *constant* in Equation 15 in the farther regions of a free jet. This indicates that the postulates adopted by these authors, and others as well, have limitations. Changes in the turbulence conditions and related slight pressure differences are important as well as *unconsidered air currents.*"

3 The idealized grid discharge of Fig. 4 is subject to a violent instability of the small jets under some conditions of geometry and flow. This would not necessarily be shown by tests over a limited range and evidently was not encountered by the authors.

4 In nonisothermal jets, the temperature has been shown by other experiments to equalize more rapidly than the velocity. Equation 30 should be used advisedly.

5 Effective temperature as used in this country involves comfort sensations which are not simple heat-transfer phenomena and which should not be oversimplified to a general linear equation without great caution. Specific limits for the approximations adopted and the basis of a free jet in a large space should be emphasized, lest erroneous applications be made.

Encouragement, however, is certainly in order for any attempt to define *draft* quantitatively, despite limitations relating to the lack of more general data.

ALFRED KOESTEL AND PHILIP HERMANN, Cleveland, Ohio (WRITTEN): During the past year, a considerable amount of co-operative research with THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was conducted at Case Institute of Technology on the subject of airflow performance from perforated panels. The results of this project are now in the form of a report and in the hands of the TAC on Air Distribution.

It will be of interest to the members of the A.S.H.V.E. to know that the theory and physical behavior of the air jets leaving a grid as pointed out by the authors is very similar to the findings of the research conducted at Case.

Tests were conducted at Case on perforated panels with a percentage of free area from 3.1 to 40.5 percent. The mechanism of airflow was found to be similar for all panels. Experimental observations indicated that the air jets from the perforations formed a homogeneous stream at a short distance from the panel face, and the cross-sectional area of this stream was dependent upon the gross panel area. The behavior of the air stream after this diffusion process was similar to that of a single large jet. Based on these physical characteristics, good correlation was obtained by equating the momentum of the individual jets to the momentum of the air stream at any distance from the panel face.

Rydberg and Norbäck describe the air discharge through a grid as follows: *The same formulas apply for single openings and for grids* because the discharged air is diffused after it leaves the grid, and the jet area is increased. The drop in velocity due to diffusion is fully compensated for by the increase in jet area.

Two dimensionless parameters based on the momentum equation, as applied to perforated panels, were used in plotting the test results.

The parameter

$$\frac{V_x}{V_o}\,\frac{1}{\sqrt{c_d\ \%\ \text{F.A.}}}\left\{\frac{\text{maximum residual velocity at the distance } X \text{ from the panel face}}{\text{outlet velocity at jet contraction}}\right\}\times$$

$$\frac{1}{\sqrt{\text{Coefficient of discharge}\left(\dfrac{\text{total hole area}}{\text{panel area}}\right)}}$$

was plotted against $\dfrac{X}{\sqrt{\text{P.A.}}}\left(\dfrac{\text{distance from panel face}}{\sqrt{\text{panel area}}}\right)$. The reciprocal, $\dfrac{1}{\sqrt{c_d\ \%\ \text{F.A.}}}$, is a diffusion parameter which corrects the residual velocity ratios, $\dfrac{V_x}{V_o}$, to values corresponding to those from uniform area outlets. A corrected residual velocity ratio of 1.18 was obtained near the perforated panel face. A value of 1.2 was found by other experimenters for conventional type ventilating outlets of 100 percent free area.

Equation 25 can be altered by proper substitution to obtain the same dimensionless parameters as found in our derivation. However, the value of K_1 equal to 5, as suggested by the authors and used in Equation 25, predicts residual velocities which are considerably higher than those found by the Case tests on perforated panels. A value of 3.5 is suggested by the test data.

Fig. 3 represents a plot which may prove useful in correlating data on stream envelopes from various types of ventilating outlets. A limited number of our experimental points for perforated panels show close agreement with the curve shown in Fig. 3.

In applying this plot for determining a stream shape, the variation of centerline velocity, V_m, with distance X, must be known from data for a given outlet. The residual velocity profile, V, can then be assumed and the envelope determined from the values of y. This method would apply to perforated panels and grills under isothermal conditions of air flow.

The theory covering the subject of air distribution and draft presented in this paper will serve as a valuable tool in developing and correlating theory and data obtained from research on air distribution projects.

G. L. TUVE, Cleveland, Ohio: There is a language difficulty in this paper introduced by the use of the term *grid*. In most cases, it is used for what we would call a *perforated panel*. In some cases, the perforations are closely spaced, giving a large, free area. Such an outlet face we would probably call a *grill*.

As you read this paper, you will find a number of things in it. Mr. Nottage mentioned one of the significant statements in this paper, mainly: the assumption that temperature and velocity equalize correspondingly. That is, if you have a stream of cold air coming from a grill and you measure the velocities or the temperatures, you may interchange the readings. In other words, if the velocity has been reduced to a lower level, is it true that you do not have to be concerned about a large temperature difference?

This is an important condition, and it is fairly closely true in most cases that we have worked with at Case Institute of Technology.

AUTHORS' CLOSURE: In the discussion of our results, two principal objections have been stated. One concerns the theory of the jet course and the other concerns our conception of draft. No theory hitherto presented on the course of free jets gives accurate coincidence with all the experimental data available, especially at large distances from the inlet opening. This may, however, as is pointed out by all experimentists concerned with these problems, be due both to the difficulty in measuring low velocities, as well as to disturbances from the surroundings. Lately, Folsom and Ferguson[18] have shown, that the turbulence, and, consequently, the mixing may die out at some distance from the inlet opening, but they also state that no measurements hitherto made make it possible to estimate the final transition point. The momentum theory agrees fairly well with experimental data, and as yet there is no really decisive evidence of the opposite, which precludes the use of this very simple way of handling the problem. As to the value of the constant K_1 Messrs. Humphreys, Nottage and Koestel all suggest lower values than 5, and this may be reasonable. The experiments by Corrsin [19] and the paper by Folsom and Ferguson[18] give values around 5 for the distant parts of the jet. Becher [20] arrived at a mean value of 6.5.

The second objection seems to be that we have oversimplified the problem of draft by introducing the quantity θ and the condition for freedom from draft. For instance, the sensitivity for draft is said to be much greater on the neck than on the wrist. This is, of course, so, as has also been verified by our measurements, but the vital point is that the reactions as to whether the air stream is felt cool, neutral, or warm in a given case coincide fairly well for all parts of the body. Mr. Humphreys suggests that the temperature difference necessary for freedom from draft should depend on the room air temperature. Our measurements have been extended over room air temperatures, ranging from 64 to 72 F, and within these limits we have found no change in the temperature difference needed. If the room air temperature is changed considerably from these values, the conditions will of course alter.

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No. 1363

PERFORMANCE OF RESIDENTIAL CHIMNEYS†

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CHIMNEY action is an important factor in the overall performance of residential heating equipment, whether the heater depends upon the chimney draft to supply the air for combustion, or only to move the products through the unit and then to exhaust them into the atmosphere. In the low-cost house particularly, because of the small size of the chimney, the chimney may be the deciding factor in the success or failure of the equipment which it serves. For this reason, it is important to establish reliable information on the performance of residential chimneys.

The information which is available in the literature includes a large number of opinions and calculations of theoretical chimney characteristics; experimental data on industrial chimneys; and also a recent extensive laboratory study of domestic chimneys made by Dill, Achenbach, and Duck[1]. However, these data are of limited value because of the large number of variables involved and because of the many questions regarding the operational characteristics of the residential chimney which still remain unanswered. The object of this paper is to seek an answer to some of these questions, inasmuch as they are of practical importance and they affect experimental techniques in chimney design.

SUMMARY AND CONCLUSIONS

The quickest and most reliable means for obtaining information on a chimney is experimental measurement, inasmuch as the large number of variables affecting the operation of a chimney make precise chimney analysis not only extremely difficult but also limited in its application.

Tests were conducted on four different types of chimneys to obtain their operational characteristics. Two of the chimneys were common brick, one was

† This study of residential chimneys has been conducted at Battelle, for Bituminous Coal Research, Inc., the objective being to obtain information on small residential chimneys for hand-fired heating equipment.

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[1] Exponent numerals refer to Bibliography.

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constructed of lightweight materials, and one was an experimental double-walled chimney designed to overcome the effects of wind.

It was found that chimney efficiency varied only slightly with the various chimneys tested, and was not greatly affected by the materials or type of construction. The cross-sectional area did have some effect on the efficiency, but only insofar as it affected the friction and the recirculation losses.

Recirculation of the flue gases within the chimney and flue pipe was found to have considerable effect on the cooling and the friction losses. The effect of recirculation decreased as the rate of flow increased.

When the flow of gases was increased beyond approximately 200 lb per hr, to simulate chimney operation with stoker equipment, the draft and efficiency

TABLE 1. NET AVAILABLE HEAT FOR VARIOUS FUELS

FUEL	NET AVAILABLE HEAT, BTU PER LB OF FLUE GAS[a]
Pittsburgh bituminous	1170
Oil	1280
Natural gas	1300

[a] These figures are based on 0 percent excess air with all fuels[2].

were found to decrease, the rate of decrease depending upon the inlet temperature. This effect was expected, since friction losses increase as the square of the rate of flow.

Tests were conducted under simulated wind conditions on the experimental double-walled chimney, and also on a commercial chimney top. Wind was found to have undesirable effects on both the double-walled chimney and the chimney ventilator. It is recognized that wind may have a deleterious effect on the operation of a short chimney because of the low available draft often provided by this type of chimney.

A method is described whereby the available chimney draft for the small residential chimney can easily be calculated with a precision of ± 10 percent.

VARIABLES AFFECTING CHIMNEY PERFORMANCE

The large number of variables involved in the operational characteristics of a chimney makes an accurate, analytical approach to the problem of chimney design almost impossible. The most important variables are the chimney-flue inlet temperature, and the rate of gas flow. These factors include the effect of velocity on friction loss, and the effect on average chimney temperature. Other variables that must be considered, because at times they play an important part in chimney operation, are: height, elevation, cross-sectional area and shape, type of interior surface, conductivity of wall, leakage, outdoor temperature and humidity, specific heat and mass of chimney, tightness of house, wind velocity and direction, adjacent structures, and design of chimney top.

Two factors which do not affect performance but which must be carefully considered are cost and fireproofness.

The type of fuel burned in the appliance has not been mentioned as being an important variable because the amount of flue gas emitted per Btu liberated does

not vary enough between various fuels to be of significance. Table 1 shows the net available heat, Btu per pound of flue gas emitted, for three common fuels.

In practice, however, the type of fuel is of consequence relative to soot forma-

FIG. 1. HEATER APPARATUS AND FLUE PIPE AT CHIMNEY NO. 1 BEFORE INSULATION WAS APPLIED TO HEATER AND PIPE

tion, because soot deposits reduce the size of the flue area and consequently increase the friction of the flue.

Construction and Safety: Chimneys other than the conventional brick type include those made of lightweight, less expensive materials, such as asbestos

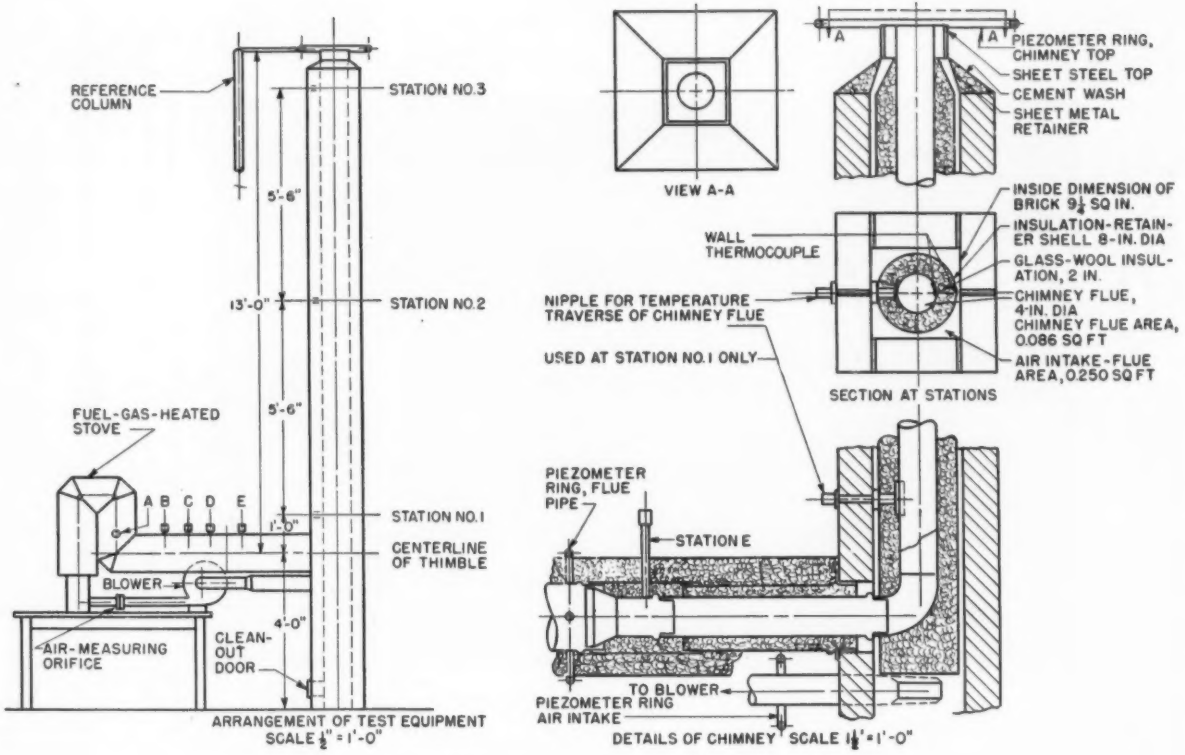

FIG. 2. DOUBLE-WALLED BRICK CHIMNEY

cement, enamelled steel insulated with rock wool, enamelled tile, or other fireproof materials. The brick chimney has the advantage of being recommended, in the Building Code[3] of the *National Board of Fire Underwriters,* for oil, gas, or coal-fired equipment, whereas few of the lightweight chimneys at the present time have been approved for all three types of fuel.

DEFINITION OF DRAFT

Draft is here defined as a *negative differential pressure;* thus, it is the absolute pressure at the thimble, less the absolute atmospheric pressure. The opposite of draft will be called *positive pressure* and will be defined as a positive *differential pressure.*

TEST EQUIPMENT

Tests were conducted on four small residential chimneys each 13 ft high above the thimble and 17 ft above the floor, and constructed as follows:

Chimney No. 1 (see Fig. 1). 8 x 8 in. clay flue lining (6¾ x 6¾ in. inside) surrounded by 4 in. thickness of common brick laid in cement mortar. Space (approximately ½ in.) between lining and brick filled with cement mortar.

Chimney No. 2 (see Fig. 1). Same as No. 1, except that flue lining sections were not cemented together and the space between lining and brick was not filled.

Chimney No. 3 was of vitreous-enamelled flue pipe, 7 in. inside diameter or a cross-sectional area of 38.48 sq in., insulated with an expanded mineral material and enclosed in an outer steel casting 11 in. in diameter. This lightweight chimney was instrumented in much the same manner as Chimney No. 2, since only comparison tests were to be conducted.

Chimney No. 4. A 4 in. I.D. steel flue surrounded by 2 in. glass wool insulation enclosed in an 8 in. diameter retainer shell and enclosed in a 4-in.-thick brick chimney 9¼ x 9¼ in. (inside). The space between the retainer shell and the inside of the brick enclosure was used as an air supply duct through which air for combustion was drawn from the top of the chimney and delivered by the fan to the fuel-gas-heated stove (see Fig. 2).

The first chimney was a conventional brick type constructed according to recommendations of the Building Code[3]. A second chimney of identical size was built for comparative purposes, in order to check the differences of opinion among heating engineers regarding the desirability of air or mortar surrounding the liner. In this chimney the liner was dropped into place and the joints were left unsealed; this is common practice, though not in accord with the Code[3].

Fig. 1 shows the two brick chimneys tested, also the combustion chamber (before insulation was applied) used to supply hot gas to the chimneys and the auxiliary equipment used in the tests. Both chimneys were constructed by a local bricklayer, who had considerable experience constructing residential chimneys. Chimney No. 1 was completely supplied with instruments to permit determination of flue-gas temperature at nine levels, surface temperatures inside and outside at these levels, and interior brick temperature at four points. The minimum of instrumentation was installed on Chimney No. 2, as it was to be used simply for comparison with the code chimney, to determine the effect of air space.

Wind Effect

Although no mention has yet been made of wind effect, it is a very important factor in the operation of a chimney. To combat this effect, P. C. Vyff, formerly associated with the project, proposed that the air for combustion be drawn from approximately the same point as that at which the flue gases were exhausted into the atmosphere. Theoretically, the wind should then have no effect on the chimney draft, because a gust of wind should pressurize equally both the combustion-air inlet and the flue-gas outlet; and then, although the absolute pressures increased, the chimney draft should remain the same. This principle was embodied in the double-walled Chimney No. 4 (see Fig. 2). The

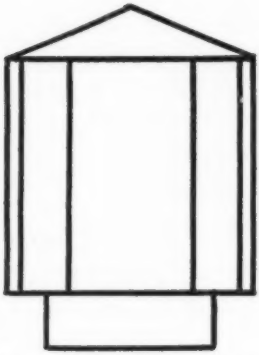

FIG. 3. CHIMNEY VENTILATOR

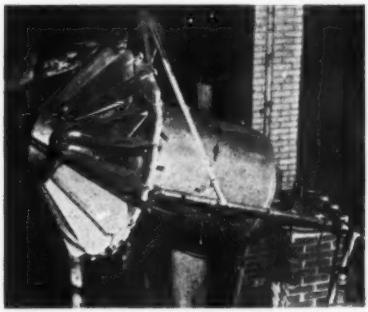

FIG. 4. BLOWER AND DOUBLE-WALLED CHIMNEY

inside diameter of the center flue was 4 in., providing a cross-sectional area of 12.56 sq in.

Fig. 3 shows a roof type ventilator which was tested under a variety of wind conditions on the common-brick Chimney No. 2.

Means for producing wind effect at the top of Chimney No. 4 are described in the following section.

TEST SETUP

A gas-fired heater was used to supply heated air and flue gases to all test chimneys. A small centrifugal blower delivered air through a measuring orifice to the base of the combustion chamber, in which were installed three gas burners. The top of this vertical, well-insulated chamber was connected to an inverted U-bend, made also of 7-in. pipe, insulated with 2 in. of rock wool and covered with a layer of aluminum foil. This inverted U-bend discharged downward into a vertical pipe, which had been incorporated to take advantage of the mixing of thermal currents inherent in this type of flow. From this vertical pipe, the flue gas was turned through a 90-deg elbow into the horizontal flue.

The temperature traverses were obtained with a high-velocity aspirating-type shielded chromel-alumel thermocouple (the precision of the temperature measurements was within ± 1 percent).

Because of the effects of adjacent equipment on the temperature of the air surrounding the chimney, an insulated, water-cooled, copper-tube reference column was installed. This column, shown in Fig. 2, was employed to eliminate any effects on the manometer lines due to unequal ambient air temperatures surrounding the chimney, an effect which was found to influence the measured draft as much as 1½ percent. Detailed calculations converted the reference to a standard of 29.92 in. of mercury and 60 F.

A large blower was constructed to test double-walled Chimney No. 4, as well as the experimental chimney top under simulated-wind conditions. Fig. 4 shows

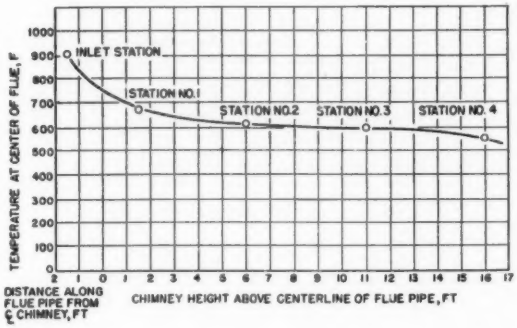

FIG. 5. VARIATION OF AVERAGE FLUE-GAS TEMPERATURE FOR A 17-FT CHIMNEY (DATA TAKEN FROM REFERENCE 1)

this blower mounted at the top of the double-walled chimney. Through the use of a specially designed mount, the angle of attack and wind direction could be set at any desired position. Extensive tests were made with various cones mounted in front of the fan to make certain that the velocity profile was uniform across the blower over the entire operating range.

TEST PROCEDURE

The same test procedure was used for all chimneys tested. Tests were made at a number of nominal inlet temperatures and at various flows. Flue traverses were made at various points along the horizontal flue and vertical chimney to determine the temperature gradient. The efficiency of the chimney was computed from the test data thus obtained, as were also the heat loss and the friction loss. The chimney efficiency, in percent of theoretical available draft, was calculated as follows:

$$\text{Efficiency} = \frac{\text{available draft}}{\text{draft calculated from the inlet temperature}} \times 100$$

The percentage of theoretical draft lost, accountable to cooling in the chimney was derived as follows:

$$\text{Cooling loss} = \frac{\text{theoretical draft} - \text{calculated draft based on average chimney temperature}}{\text{ideal draft}} \times 100$$

The percentage of theoretical draft lost because of apparent friction in the chimney was derived as follows:

$$\text{Apparent friction} = \frac{\text{calculated draft} - \text{available draft}}{\text{theoretical draft}} \times 100$$

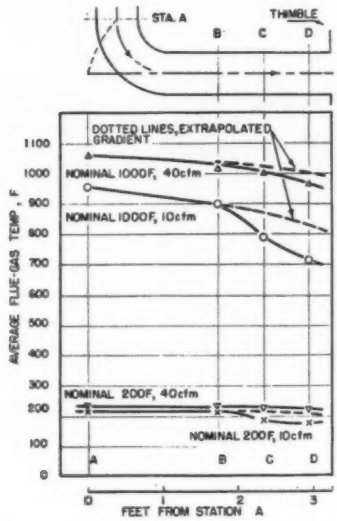

FIG. 6. VARIATION OF AVERAGE GAS TEMPERATURE ALONG FLUE PIPE

Tests were made on the double-walled chimney and the ventilator under a variety of wind conditions. With the temperature, flow, and direction held constant, the angle of attack was varied to see what effect wind at various vertical angles would have on the chimney draft. Tests were also conducted, in the same manner, to determine what effect wind direction had upon the chimney draft.

CHIMNEY CHARACTERISTICS

Recirculation and Temperature Gradient

Two reasons for two hitherto unexplained discrepancies (effect of temperature drop at chimney entrance and effect of friction loss) that had appeared in data from previous chimney investigations[1] and which have also appeared in the authors' data, were disclosed from tests that were conducted on Chimney No. 1.

The *first* discrepancy pertained to the temperature drop at the point where the flue pipe entered the chimney proper.

Fig. 5 shows the temperature gradient in a chimney tested by Dill[1]. The decided temperature drop at the thimble is shown clearly by this figure.

Figs. 6 and 7 show the temperature gradient and the vertical temperature profiles for several test conditions as found by a temperature survey made in the horizontal flue of Chimney No. 1. This was made to determine the type of flow occurring in the duct. From these figures can also be noted the sharp drop in temperature at the chimney inlet, and the variation in the temperature profile. Weske[4] has shown that for elbows of more than 30-deg angle of bend and of small radius ratio, separation does occur, frequently accompanied by a reversal of

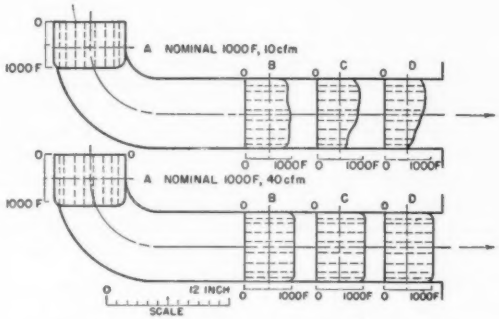

FIG. 7. TEMPERATURE PROFILES IN INSULATED FLUE PIPE SHOWING EFFECTS OF INLET TEMPERATURE AND FLOW RATE ON RECIRCULATION IN CHIMNEY

flow; whereas a region of eddying flow near the inside wall of the curved ducts is encountered in all cases. The chimney-temperature surveys and the work done by Weske would indicate that undoubtedly recirculation of the flue gas is taking place; hence the discrepancy in the flue-gas temperature at the thimble.

A sudden change in cross-sectional area may at times be of importance. An enlargement of the horizontal flue or vertical chimney flue may result in added reversal of flow *(recirculation)* because of the decrease in the velocity of the gases; while a decrease in cross-sectional area would cause an increase in the friction losses and possibly some recirculation before the restriction. This latter effect was noticed with the double-walled chimney where a decrease in cross-sectional area in the horizontal flue caused an uneven temperature gradient before the restriction, a condition which could only be attributed to internal turbulence.

Fig. 8 shows this uneven temperature gradient in the horizontal flue. It may be noted that the gases flow from the horizontal flue into Chimney No. 4 through a 90-deg elbow, while in Chimneys 1, 2, and 3, as in most chimneys, the horizontal flue is connected to the chimney by a tee. This latter arrangement in all probability would increase the recirculation losses.

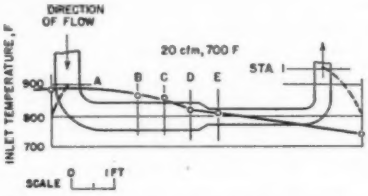

FIG. 8. VARIATION IN HORIZONTAL FLUE TEMPERATURE WITH REDUCTION IN CROSS-SECTIONAL AREA

Recirculation and Friction

The *second* important discrepancy was related to the effect of the friction loss, in the chimney, on chimney performance. It was found from previously published data on friction factors in roughened pipes[5] that the friction factor would, regardless of roughness, approach a maximum of 0.016, a value that is also used for chimneys in the HEATING, VENTILATING, AIR CONDITIONING GUIDE. But even with a friction factor of 0.016, the actual friction loss was much larger in the majority of tests than the calculated theoretical friction loss. When recirculation occurs, it affects the friction losses; therefore, it would

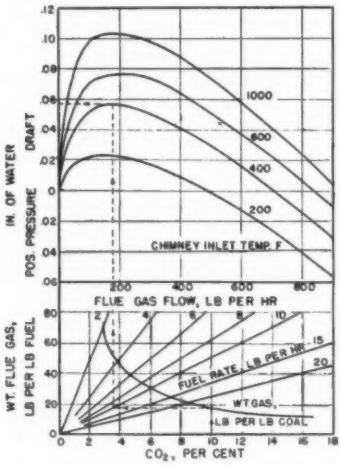

FIG. 9. GRAPHICAL EVALUATION OF THE AVAILABLE CHIMNEY DRAFT FROM PERCENTAGE CO_2, FUEL RATE, AND CHIMNEY-INLET TEMPERATURE (FOR CHIMNEY NO. 1, H.V. BITUMINOUS COAL, 29.92 IN. HG, 60 F)

appear that much of the apparent friction loss actually results from eddy and recirculation losses. The calculated friction is extremely small, and depends wholly upon the accuracy of the test data. It is believed that the test data, prin-

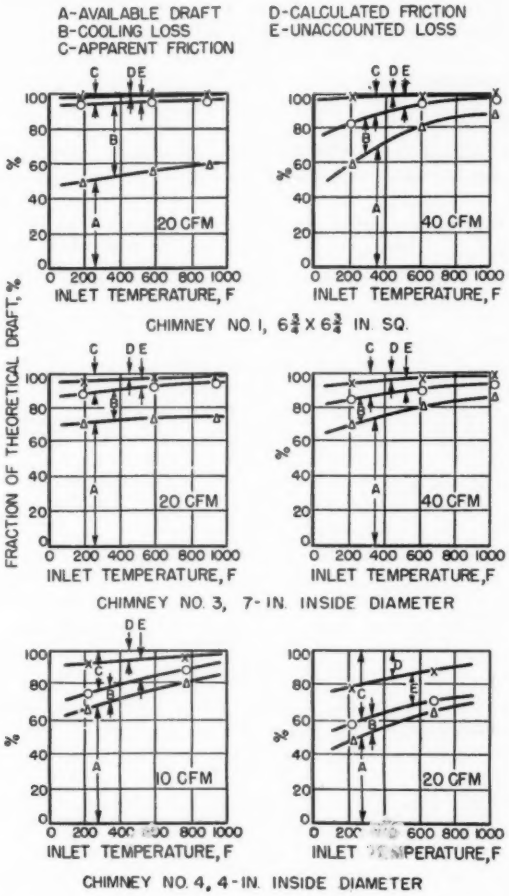

FIG. 10. DRAFT BALANCE PLOTTED AGAINST TEMPERATURE FOR THREE TYPES OF CHIMNEYS

cipally the temperature measurements, are not precise enough to warrant accurate calculations of the friction loss. An error of a few degrees in temperature measurement (and this error might again be blamed on recirculation) would have a considerable effect on the calculated friction loss.

Recirculation of the flue gases was found throughout the entire length of the chimney at low flows. This was verified by observing the flow of gases into the chimney top by the striation or shadowgraph method, in which it was noted that at extremely low rates of flow there were periodically small amounts of outside air which flowed into the chimney top, while the gases surrounding the chimney momentarily shifted to one side. However, as it was not observed at the higher flow rates, it is thought that in the practical range of flows this effect does not occur.

Correction for Recirculation

It has been pointed out that because of recirculation there is an appreciable drop in temperature where the flue gases enter the chimney; hence it can be

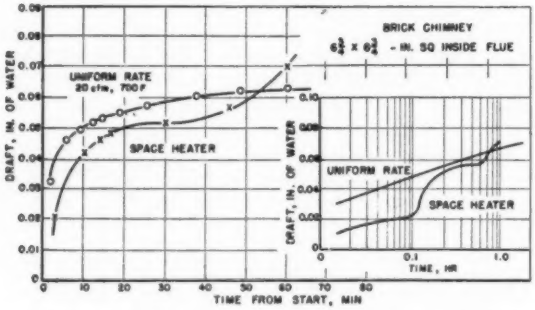

FIG. 11. COMPARISON OF DRAFT PICK-UP RATES (STANDARD 13-FT BRICK CHIMNEY SUPPLIED WITH HOT GAS AT 20 CFM AT UNIFORM RATE, AND THE SAME CHIMNEY CONNECTED TO A SMALL COAL-FIRED SPACE HEATER)

seen that the chimney-inlet temperature obtained from a test survey and based on the extrapolated temperature gradient would not be a true indication of the chimney-inlet temperature. This chimney-inlet temperature would be some function of the recirculation, which is a factor that would be almost impossible to analyze.

For the first few tests, the theoretical temperature gradient of the horizontal flue was calculated from heat-transfer data. This new curve was extrapolated to the thimble of the chimney, and the resulting temperature was assumed to be the true chimney-inlet temperature. Fig. 6 shows this theoretical curve in dotted lines, but for the majority of tests the theoretical curve was used only as a guide, because a curve could easily be drawn that approached the theoretical with an error of only a few degrees.

Effects of Construction

The effects of the methods of construction on chimney efficiency were analyzed by comparing the results of the performance tests of Chimneys No. 1 and No. 2, and the comparison showed that the presence of an air space instead of

mortar surrounding the liner in the brick chimney had no effect at all on the chimney operational characteristics. Moreover, there seems to be no appreciable insulating value in the air space. From a mechanical standpoint, therefore, the standard grouted-liner construction is preferable since it holds the liner in place when it becomes cracked.

Available Draft

Fig. 9 is a graphical presentation of the available chimney draft over a wide range of flows for a standard 13-ft brick chimney with an 8 x 8-in. (nominal-size) flue liner. Using performance values of the heating unit at full rating, it is possible to determine from this graph whether the chimney is adequate for the particular heating unit attached to it. By entering the chart at percentage CO_2, moving vertically to the curve representing pounds of flue gas per pound of fuel, then horizontally to the fuel rate and vertically to the chimney inlet temperature, the available draft may easily be determined.

The use of this chart can best be explained by a numerical example: 10 percent CO_2, fuel rate 10 lb per hr, and a chimney inlet temperature of 400 F. These conditions, it can be seen, will give an available chimney draft of 0.057 in. water for a 13 ft chimney. The margin of capacity is obtained by comparing this value with the rated draft of the appliance.

With a chimney attached to natural-draft equipment, the draft obtained is an important factor in the correct operation of the heating equipment; but with a stoker which is under forced draft, draft is important only to prevent efflux of gas from the equipment.

It may be noted that for flow rates above approximately 200 lb per hr that the draft and the efficiency decrease. This effect is to be expected, as the friction loss in a duct increases as the square of the rate of flow.

Efficiency

Fig. 10 shows the comparative efficiencies of Chimneys Nos. 1, 3, and 4: (A) is the fraction of the theoretical draft obtained as available draft, (B) is the fraction of the theoretical draft lost by cooling, (C) is the fraction of the theoretical draft lost because of apparent friction, and (D) is the fraction of the theoretical draft lost from calculated friction. (E) The difference between the fraction of the theoretical draft lost because of apparent friction and the calculated friction is an unaccounted-for loss. It can be seen that the average available draft lies between 70 and 80 percent of the theoretical draft for the average operating conditions, although Chimney No. 3, a lightweight insulated chimney, has a higher efficiency at low temperatures and flow rates than the standard brick chimney.

Pickup

With lightweight Chimney No. 3 there is a slightly faster rate of pickup than for a brick chimney, but the pickup rate of any standard chimney considered by itself is probably greater during the kindling period than is the combustion pickup of most coal-fired heating equipment. A possible exception to this rule would be with stokers, as these operate under forced draft and their pickup rate can be very high.

Fig. 11 is a comparison of two typical pickup curves, one for a brick chimney

with the laboratory heater, and one for a small hand-fired space heater attached to the same chimney. It can be noted that the combustion pickup of the space heater and chimney is lower than the pickup when the chimney is supplied with hot gas from the start; hence the advantage of special pickup merits of insulated chimneys is small with natural-draft equipment.

WIND EFFECT ON THE CHIMNEY TOP

The varying effects of wind upon a chimney have always been a troublesome problem, inasmuch as air flowing around and over a building creates many eddies and vortices which may affect the chimney draft to a considerable extent.

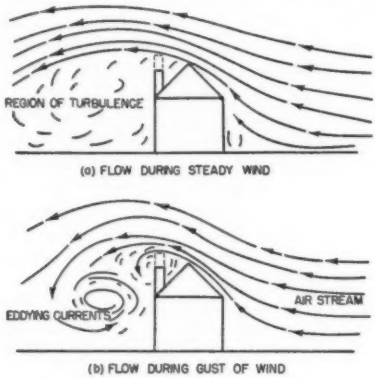

FIG. 12. AIR FLOW OVER A BUILDING AND CHIMNEY

These effects may either be from ram pressure upon the chimney itself, or from the increase or decrease in pressure around the house because of the changing velocity of the air stream as it flows over the dwelling. Fig. 12, (*a*) and (*b*), shows the air flow over a small house and chimney for a gust of wind, and also for wind at a constant velocity[6]. To make the illustration clearer, the chimney height is shown below the ridge, and in dotted lines above the ridge of the house. It can be seen that it is important to build the chimney a few feet higher than the ridge to avoid the region of turbulence. It is frequently recommended that the chimney should be a foot or more higher than the ridge of the house, although exact dimensions cannot be given, because of the many varying physical conditions and the everchanging wind velocities. A chimney top only as high as or below the ridge may well be in the region of turbulence, and the result may be a marked decrease in draft.

A theoretical example of the increase in pressure at the chimney top due to change of velocity head to static pressure, *ram pressure*, at this point, was computed for an air-stream angle (angle of attack) of 30 deg above the horizontal plane of the chimney top, and a wind velocity of 40 mph. The increase in pressure under these conditions amounted to over 0.100 in. of water. For a small

residential-type chimney this increase in pressure at the chimney top would probably be greater than the available draft, as an average draft of approximately 0.09 in. of water can be expected for a chimney only 15 ft in height. From this example it can be seen that for the smaller chimneys that are used for low-cost housing the ram effects of wind may become very serious.

The suction and pressure effects around the house which are due to the change in velocity of the air stream are almost impossible to evaluate with any degree of accuracy, but they too will have damaging effects on the performance of the chimney. It should be emphasized again that a chimney built above the ridge of the house will minimize these disturbances considerably.

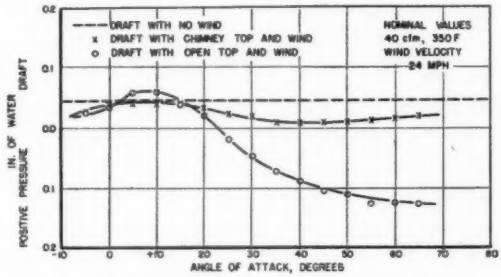

FIG. 13. DRAFT VARIATION WITH CHANGE IN ANGLE OF WIND ATTACK (UNDER SIMULATED-WIND CONDITIONS WITH CHIMNEY TOP)

TESTS UNDER SIMULATED-WIND CONDITIONS

The Double-Walled Chimney: The double-walled chimney, which was specifically designed to overcome wind effects, did not function as was hoped; wind had a marked effect upon the draft regardless of what measures were taken to counteract it. Compared to other chimneys previously examined, the efficiency of this chimney is favorable, which is probably due to the low heat loss. It appears that this type of chimney is no better than the conventional chimney in overcoming the effects of wind upon the draft, but it could be useful in homes where the tightness of the house would restrict the air for combustion, creating an adverse effect on the chimney draft. With this arrangement, the house could be perfectly tight without affecting the operation of the heating unit.

The Chimney Ventilator: Fig. 13 shows the results of the tests made on Chimney No. 2 under simulated-wind conditions both with and without the ventilator shown in Fig. 3. The draft variation is plotted against the air-stream angle of attack for a wind velocity of 24 mph.

A chimney ventilator, if properly installed and maintained, will in the majority of cases, counteract the wind to a considerable extent. There are cases, for example, a very tight house, where a chimney ventilator will not help; hence, it is not a *cure all* for all chimneys having poor draft, or those connected to equipment which may produce even a small amount of soot, the accumulation

of which over a period of time might affect the operation of the ventilator by clogging any small passages.

DETERMINATION OF AVAILABLE CHIMNEY DRAFT

One of the most important problems in the design and construction of residential chimneys involves the determination of the available draft for any small chimney. The available draft will be some function of the ideal draft, and the losses from conduction, friction, and recirculation. It can be seen that a rigidly complete solution for determining the available draft would be extremely unwieldy and subject to inaccuracies which would be due to the assumptions of

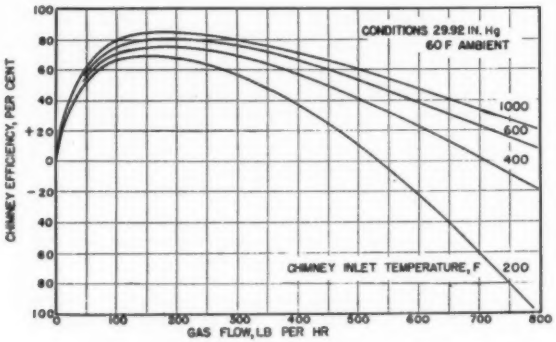

FIG. 14. CURVE OF EFFECT OF FLOW ON EFFICIENCY DERIVED FROM TEMPERATURE PLOTS (CHIMNEY NO. 1)

various heat-transfer coefficients, friction factors, and other variables; hence, a reliable, approximate solution to this problem is desirable.

The available chimney draft may be written in the following manner,

$$D_a = \eta D_t \quad \ldots \quad (1)$$

where

D_t = theoretical chimney draft, based on inlet temperature.

η = chimney efficiency.

It has been found from the test data obtained in this investigation that for a small residential chimney 10 to 25 ft in height, and with a cross-sectional area of 35 to 55 sq in., the chimney efficiency will vary only within ± 10 percent for chimneys at the same flow and temperature conditions regardless of friction and cooling losses; therefore, the product of the theoretical draft and the efficiency obtained from a previously tested chimney, at the desired flow and temperature conditions, will give an approximate solution for the unknown available draft. It may be expressed algebraically as:

$$D_a \cong \eta_c D_t \quad \ldots \quad (2)$$

where

D_t = theoretical chimney draft, inches of water.

η_c = chimney efficiency taken from Fig. 14, at the desired conditions of temperature and flow.

Fig. 14 is a plot of chimney-efficiency test data obtained over a wide range of flows for common brick Chimney No. 1.

The theoretical chimney draft, in inches of water, may be computed from the following equation:

$$D_t = 0.2554\, BH \left[\frac{1}{T_o} - \frac{1}{T_c}\right] \quad \ldots\ldots\ldots\ldots \quad (3)$$

where

B = barometric pressure, inches of mercury.
H = chimney height, feet.
T_c = chimney inlet temperature, Fahrenheit absolute.
T_o = ambient air temperature, Fahrenheit absolute.

The slight change in efficiency due to the variation of the cooling and friction losses with changes in height, cross-sectional area, and materials, is disregarded in this method. This is done to simplify the procedure as much as possible, and to make a workable solution which can easily be used. It must be remembered, however, that it is only an approximate method for small residential chimneys with a height from 10 to 25 ft, and a cross-sectional area from 35 to 55 sq in.

An example of the use of this method may be had by comparing it with some actual test results. The following example is given: chimney height, 13 ft; inside diameter, 7 in. (area 38.5 sq in.); inlet temperature, 205 F; flow, 270 lb per hr, ambient air, 60 F, and barometric pressure, 29.92 in. Hg.

The theoretical draft as calculated from the draft equations:

$$D_t = 0.2554 \times 29.92 \times 13 \left[\frac{1}{520} - \frac{1}{665}\right]$$
$$= 0.0397 \text{ in. of water.}$$

From the curve in Fig. 14 the chimney efficiency may be found:

$$\eta_c = 0.60.$$

Substituting η_c and D_t in Equation 2 and calculating,

$$D_a = 0.60 \times 0.0397$$
$$= 0.0238 \text{ in. of water.}$$

The actual test results corrected to 29.92 in. Hg at 60 F gave,

$$D_a = 0.0239 \text{ in. of water,}$$
$$\text{Error} = 0.4 \text{ percent.}$$

The reliability of this method of determining chimney draft was further indicated when checking it with the results obtained from Dill's test data[1]. Calculations were made on a 22.3-ft chimney (inside dimensions 7.25 x 7.25 in.) correcting the available draft that he observed to 60 F. The available draft that Dill recorded compared within 6.9 percent of the draft that was calculated. This percent error is well within the allowed range.

If the percent CO_2 and fuel rate are the only known factors for the determination of the available draft for the unknown chimney, then the flue-gas flow may be determined from the lower plot of Fig. 9, and the problem worked as just described.

It should also be noted that at the higher flows which simulate operation with stoker equipment, the error due to friction losses will increase as the flow increases, but a reliable determination of the available draft may still be had at only a slight increase in error.

ACKNOWLEDGMENTS

The contributions to this investigation of P. C. Vyff, F. B. Terrill, and W. T. Brown, are gratefully acknowledged.

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DISCUSSION

P. R. ACHENBACH, Washington, D. C. (WRITTEN): This paper presents interesting and useful information on recirculation in chimneys, wind effects, chimney caps, and chimney efficiency. There is general agreement between the test results on chimney draft and chimney efficiency reported in this paper and those obtained at the National Bureau of Standards. The reference column used for draft measurements and the indoor installation of the chimneys probably provided better control of test conditions and greater precision than could be obtained with chimneys extending out-of-doors.

Recirculation and eddy currents undoubtedly affect the observed temperature drop at the entrance to a chimney and the actual draft loss due to friction. However, some cooling of the gases would occur at the chimney inlet due to conduction downward through the brick in the base of a chimney that would be independent of down-drafts or eddy currents. Eddy currents near the sidewalls of a chimney could be predicted from an analysis of the pressure and temperature gradients in a chimney. Since the gases are cooled near the sidewalls, the weight of the columns of gases in an annular space near the sidewalls would be greater than at the center. Consequently, in the lower parts of the chimney the draft would theoretically be greater in the center than near the walls. This difference in pressure would cause movement of the gas from the sidewall toward the center of the gas stream in the lower part of the chimney and probably cause eddy currents near the sidewalls throughout the chimney. Such an analysis would suggest that a well-insulated chimney would have less eddy currents and therefore less friction.

Table 1 and discussion relating thereto indicate that the heat liberated in a furnace per unit weight of flue gas produced is about the same for coal, oil, and gas. This is somewhat misleading. Approximately 20 percent excess air is used for combustion of gas, 50 percent for fuel oil, and perhaps 100 percent for coal. The values in Table 1 then become

FUEL	NET AVAILABLE HEAT, BTU PER LB OF FLUE GAS
Pittsburgh bituminous	585
Oil	853
Gas	1083

That is, coal may produce nearly twice as much flue gas per unit of heat liberated as gas.

In the last sentence under the heading *Efficiency,* it is stated in referring to Fig. 10, *It can be seen that the average available draft for the average operating conditions. . . .* The meaning of *average operating conditions* should be made clear because Fig. 10 does not show any such consistency in chimney efficiency. Chimney No. 1 did not attain 70 percent efficiency for any inlet temperature at 20 cfm flow rate which is about the amount of flue gas produced when an oil space heater burns ½ gal of oil per hour. At 40 cfm both Chimneys Nos. 1 and 3 have efficiencies of 70 to 80 percent in the range of inlet temperature from about 350 to 600 F only.

It is believed that the data presented on pickup are too meager to warrant the conclusions drawn. Pickup of the chimney draft and pickup of the fire are treated as independent phenomena when actually they are dependent on each other. The paper does not show any comparison of pickup between two chimneys of different construction, but rather two tests of the same chimney under different conditions.

Pickup data on different chimneys obtained at the National Bureau of Standards indicate draft differences of 0.02 in. W.G. or more (25 to 30 percent difference) for identical conditions of pickup where the inlet flue gas temperatures were increased at a uniform rate from 250 to 1000 F in intervals of 5 min, 20 min, and 1 hr. These tests also showed that the time required to raise the draft on a pot-type oil burner to 0.06 in. W. G. ranged from 1½ to 4 min, and the time to raise the draft to 0.07 in. W.G. ranged from 7 to 39 min after turning the burner on high fire. These results indicate that the difference in pickup characteristics of different chimneys may be very significant for natural draft heating equipment and especially so for short chimneys.

It should be pointed out in the text of the paper that the curves in Fig. 14 can be used to determine the efficiency of a chimney for ambient temperatures near 60 F only. The efficiency of a chimney varies with the ambient temperature. The curves in Fig. 14 predict drafts within ±10 percent of the values observed in the 7 in. round and 7 x 7 in. square chimneys 15 ft in height tested at the National Bureau of Standards when these values are corrected for an ambient temperature of 60 F. However, they do not show the relationship between available draft and theoretical draft at 32 F ambient temperature or any other ambient differing from 60 F too widely. The deviation of the observed drafts in the chimneys with 7 x 11 in. liners from those predicted by Fig. 14 are of the order of ±15 percent for heights of 15 ft.

Tests of a brick chimney with 7 x 7 in. liner conducted at the National Bureau of Standards in 1941 for which some of the results are contained in THE GUIDE 1948 indicated drafts deviating as much as 25 percent from those predicted by Fig. 14 for heights of 27.5 ft. For some of the shorter chimneys the correlation was within ±10 percent. Therefore, in our opinion it is doubtful whether Fig. 14 would give results within ±10 percent for heights up to 30 ft because of the greater cooling of the gases in taller chimneys.

CARL H. FLINK, New York, N. Y.: I notice that Mr. Achenbach in his discussion referred to 20 percent excess air as the amount that would be used with a gas appliance. I am quite aware that one can adjust a gas appliance to 20 percent excess air and have safe combustion, but I doubt whether any gas-fired heating appliances are designed for operation with 20 percent excess air. I think the quantity is more nearly 35 percent if one is going to be sure that the appliance will be able to pass all *A.G.A.* approval tests.

G. V. PARMELEE, Cleveland, Ohio: I would like to see a little more information on friction losses in this paper, for two reasons. First, we have most of our information on friction losses for isothermal flow; that is, the air is at a constant uniform temperature. This study seems to offer some opportunity to get good friction data on non-isothermal flow. I do not know exactly how the comparisons between THE GUIDE data for friction loss and the figures given in this paper were made. I would like to obtain more information on whether the entrance and exist losses were deducted from the total friction loss in the chimney.

J. H. CLARKE, Whiting, Ind.: I was interested in the graph of Fig. 13 with reference to the ventilator cap which was tested. I can understand that one would probably get some *ram* effect, as the authors call it, or positive pressure on the building. I would not expect, however, that this would effect an induced draft type of ventilator, unless the ventilator was located *on* the roof, rather than on a chimney above the roof. I wonder what accounts for the apparent wide effect of this positive pressure with respect to the roof, to the point where it would affect the cap itself. The curves of Fig. 13 probably represent tests where the cap was located below the peak of the roof, but I have had some experience with ventilators in conditions which would simulate or possibly be much worse than the conditions represented by the tests conducted by the authors. I have never detected any indication of positive pressure or back pressure on this type of ventilator.

During the war we ran a considerable number of tests on frigates, sea-going tugs, and cargo ships where such ventilators were frequently installed. They were located in all sorts of places, many of which certainly would not be advocated by the designers of the equipment. To the best of my knowledge we never got any back draft on those ventilators except in some cases where the internal part of the ship was subjected to a negative pressure. Of course, in some cases this negative pressure would be much more than the ventilator head was capable of operating against. I would like to ask the authors if there is any further explanation in the experimental data which might clarify the positive pressure conditions indicated by the curves of Fig. 13.

MR. ACHENBACH: I believe I got my information partly from THE GUIDE 1948. I was checking the data in THE GUIDE and noted that it states that the average excess air is likely to be 30 to 35 percent for gas burners, but it may be as low as 20 percent.

AUTHORS' CLOSURE: We wish to thank Mr. Achenbach for his excellent criticism of our work and to remark that it is seldom that one can make so direct a comparison of one's test data with that of others and find that the results are quite comparable.

Mr. Achenbach's explanation of the phenomenon of recirculation is exceedingly interesting. Undoubtedly, these effects are taking place in the chimney to some degree. His conclusions that a well-insulated chimney would have less eddy currents, and therefore less friction, are actually borne out by some of our unpublished data. We have found that in an uninsulated, air-cooled chimney there may be as much as 60 percent greater apparent friction than there is in an insulated brick chimney; this effect seems to be diminished as the temperature is lowered, as would be expected.

We realize that the percentage of excess air will vary with the type of fuel fired. However, we still maintain that the percentage of excess air in the flue gas during combustion of various fuels is unimportant; our interest is only in the net available

heat per pound of flue gas. It is true that the percentage of excess air will be higher than what we have stated for the three fuels during combustion, and that these various degrees of excess air will affect the amount of flue gas liberated. The majority of the curves in the report were plotted with one of the variables being flow rate; hence, it makes no difference at what percentage of excess air the fuel is consumed. The performance of the chimney is concerned only with the quantity and the temperature of the gases flowing in the flue.

We stated that the average available draft was between 70 and 80 percent of the theoretical draft for the average operating conditions. This so-called efficiency, as can be seen from Fig. 10, at the very low flow rates is not exactly 70 percent but in the majority of figures it is either in or above the range which we listed.

Admittedly, the results on pickup are laboratory results and should, of course, be interpreted as such. We are glad to learn of additional information on pickup as performed by the National Bureau of Standards; they have obviously gone into this problem deeper than we have attempted.

If values of draft are required at ambient temperatures lower than 60 F, they obviously must be corrected. A temperature of 60 F was chosen as a standard because most of the laboratory tests were conducted at an ambient temperature between 60 and 70 F. These data at 60 F and at 29.92 in. Hg can be corrected to any desired ambient temperature by the following relation:

$$O_2 - O_1 = S_2 - S_1.$$

where

S_1 = theoretical draft at 60 F and at 29.92 in. Hg.
S_2 = theoretical draft at desired temperature and barometer reading.
O_1 = observed draft at 60 F and at 29.92 in. Hg.
O_2 = observed draft at desired temperature and barometer reading.

The theoretical draft is calculated from the theoretical draft equation, Equation 3.

Our method of calculating the available chimney draft is admittedly an approximate solution to a very difficult problem and should be treated as such. Up to this time, there has not been formulated for this problem a clear solution that could easily be used by the average person interested in chimney design. The data that have been published have been for individual chimneys, of which there are hundreds, and the raw data presented in tables have often times been confusing; hence, an attempt was made to correlate the results in a form that would be of greater use.

We stated that the method is only valid for chimneys with a cross-sectional area from 35 to 55 sq in. and from 10 to 25 ft in height. We were well aware that the drafts of the chimney that Mr. Achenbach mentioned did not correlate exactly with those of our method, but that chimney had a height of 27.6 ft; hence, one would expect some discrepancy.

We wish we could give more information to Mr. Parmelee on the matter of friction loss. It would be of interest, and no doubt clear up many hazy facts on chimney characteristics, if we could eliminate the recirculation effect so as to separate it from friction for the non-isothermal flow. Mr. Parmelee is right in that for isothermal conditions one is apt to obtain different friction values than for non-isothermal conditions, although an analysis from an adiabatic standpoint showed that because of the low velocities the flow more nearly approached an isothermal condition. We are making the assumption in this problem that for all practical purposes the flow is isothermal, and that the difference between our measured pressure loss and the calculated isothermal friction is entirely offset by recirculation. This, of course, will create some degree of error in our assumption, but we feel that it will not be of a serious order.

The theoretical friction losses were calculated by the method that is recommended in THE GUIDE. To the question of whether exit and entrance losses were taken into

consideration in the overall computation of pressure loss, the answer is that they were taken into consideration.

Mr. Clarke asked how, in view of his experience with the performance of ventilators on shipboard, we could have obtained a positive pressure with a chimney ventilator under simulated wind conditions. We must conclude that we had conditions which may not have been encountered in his experience, as we were venting highly heated gases through a relatively small flue. We have not gone any further into the explanation of this and would certainly welcome any more data on a chimney and ventilator used in this fashion.

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SEMI-ANNUAL MEETING, 1949

MINNEAPOLIS, MINN.

FOUR hundred members and guests came to Minneapolis for the Semi-Annual Meeting of the Society, June 20-22, 1949. Ten papers were presented and discussed at three well-attended technical sessions.

FIRST SESSION—MONDAY, JUNE 20, 10:00 A.M.

Pres. A. E. Stacey, Jr., Syracuse, N. Y., called the 1949 Semi-Annual Meeting to order at 10:00 a.m. in the Nicollet Hotel, Minneapolis. John E. Haines, chairman of the Committee on Arrangements, cordially welcomed the members and guests.

President Stacey then called upon Secretary A. V. Hutchinson, New York, N. Y., who read the amendment proposed to **Article B-XI—Funds,** *Section 7,* of the Society's By-Laws.

Article B-XI—Funds

Section 7. A Society Reserve Fund shall be created into which all admission fees and such other moneys as the Council may direct shall be placed until the fund totals a sum equal to fifteen dollars ($15.00) per member, *or fifty thousand dollars ($50,000) whichever is smaller.* This Reserve Fund is to be used only in cases of emergency when current Society revenues are not sufficient to pay necessary expenses. Withdrawals not to exceed twenty percent (20%) of the fund in any calendar year may be authorized by the Society at any meeting provided it has been recommended by the Council. Investments shall be made in accordance with **Article B-XI,** *Section 1.* Interest earned on the Reserve Fund shall be added to current Society income.

To be Amended as Follows:

Section 7. A Society Reserve Fund shall be created into which all admission fees and such other moneys as the Council may direct shall be placed until the fund totals a sum equal to fifteen dollars ($15.00) per member. This Reserve Fund is to be used only in cases of emergency when current Society revenues are not sufficient to pay necessary expenses. Withdrawals not to exceed twenty percent (20%) of the fund in any calendar year may be authorized by the Society at any meeting, provided it has been recommended by the Council. Investments shall be made in accordance with **Article B-XI,** *Section 1.* Interest earned on the Reserve Fund shall be added to current Society income.

Adoption of the Amendment was moved by Merrill F. Blankin, Philadelphia, Pa., and seconded by L. E. Seeley, Durham, N. H. Dean Seeley explained that the purpose of the Amendment was to increase the Society Reserve Fund by removing the limitation of $50,000.00 which had been established when the Society had less than one-half of the present membership.

The Amendment was adopted unanimously.

President Stacey called upon N. A. Buckley, special assistant in Mechanical Engineering, University of Illinois, Urbana, Ill., who presented the paper,

RESULTS OF OPINION SURVEY ON A.S.H.V.E. RESEARCH PROGRAM

	TOTAL VOTES	WOULD LIKE TO SEE MORE EMPHASIS IN FUTURE PROGRAMS	ARE SATISFIED WITH PRESENT PROGRAM	SUGGEST LESS EMPHASIS IN FUTURE PROGRAMS	ORDER OF INTEREST BASED ON COL. 2
AIR DISTRIBUTION					
Friction of fittings	890	204	517	169	31
Performance of outlets	909	418	411	80	7
and intakes	852	336	427	89	16
Ventilation jets	824	284	441	99	24
Effect of approach conditions	814	190	519	105	32
Air flow measurement	891	306	486	99	21
AIR PURIFICATION					
Air cleaning in ventilation	911	374	475	62	11
Dust control	864	327	476	61	18
Atmospheric pollution	865	318	469	78	19
Air sterilization	910	462	380	68	6
Industrial and exhaust ventilation	871	368	418	85	12
Odors and odor control	893	467	361	65	5
COMFORT AND HEALTH					
Influence of humidity	926	410	446	70	9
Influence of radiation	901	404	444	53	10
Influence of air purity	894	330	492	72	17
Rapid environmental changes	897	351	460	86	14
Panel or radiant heating	1016	542	390	84	1
Panel or radiant cooling	938	522	315	101	2
Heat pump	956	496	318	142	3
Heating load factors	896	287	527	82	22
Cooling load factors	867	307	494	66	20
Heat transmission through building materials	899	246	539	114	30
Heat capacity of building materials	869	286	495	88	23
Infiltration	894	341	490	63	15
Solar radiation	907	413	436	58	8
Insulation	865	264	526	75	29
Water vapor migration	852	361	423	68	13
Weather design conditions	858	283	486	89	25
Fuels and combustion	869	273	484	112	27
Chimneys and draft	869	269	463	137	28
Moisture control by sorbents	831	274	465	92	26
Noise control	897	470	379	48	4
Psychrometric charts	837	166	543	128	33

Determining Performance of a Blower Used with a Furnace, by Mr. Buckley and R. W. Roose (see Chapter 1365). Mr. Buckley gave a resumé of the paper, together with the significant conclusions derived.

Air Streams from Perforated Panels, by Alfred Koestel, Philip Hermann and G. L. Tuve (see Chapter 1366), was presented by Mr. Koestel, instructor, Mechanical Engineering, Case Institute of Technology, Cleveland, Ohio, who illustrated his resumé of the paper by means of slides.

Pressure Loss and Air Flow Characteristics of a Box Plenum, by S. F. Gilman, R. J. Martin, W. R. Hedrick, and S. Konzo (see Chapter 1367), was

presented by Mr. Gilman, instructor and special research assistant, University of Illinois, Urbana, Ill.

The meeting adjourned at 12:00 noon.

SECOND SESSION—TUESDAY, JUNE 21, 9:30 A.M.

First Vice Pres. Lester T. Avery called the meeting to order at 9:30 a.m., and introduced L. N. Hunter, Johnstown, Pa., chairman of the Committee on Research.

To aid in developing a research program that will be of maximum benefit to the membership, Mr. Hunter stated that the Committee wanted to know:

1. To what extent the membership, old and new, understood the purpose of Society research and the kind of research permissible under the Constitution.
2. What kind of research the membership really wanted.
3. Which parts of the current program were of most interest.
4. How the membership felt about the present methods of getting into their hands the information developed as a result of Society research.

Mr. Hunter also stated that nearly 1300 replies were received, in response to a letter sent to the membership, and that a committee consisting of John W. James, vice chairman of the Committee on Research, R. C. Cross, and M. K. Fahnestock, analyzed the returns which were tabulated and shown in the accompanying table.

First Vice President Avery introduced E. W. Guernsey, director of research, Consolidated Gas Electric Light & Power Co., Baltimore, Md., who presented the paper, Earth as a Heat Source or Storage Medium for the Heat Pump, by Dr. Guernsey, P. L. Betz, and N. H. Skau (see Chapter 1368).

J. Donald Kroeker, consulting engineer, Portland, Ore., presented the paper, Heat Pump Results in Equitable Building, which he prepared with R. C. Chewning and C. E. Graham, as co-authors (see Chapter 1369) and which was a review of the results obtained in the operation of the heat pump installation in the Equitable Building, Portland, Ore.

J. V. Borry, Minneapolis, Minn., presented the paper, Ground Temperatures as Affected by Weather Conditions, by A. B. Algren, as Professor Algren was unavoidably absent because of illness (see Chapter 1370).

THIRD SESSION—WEDNESDAY, JUNE 22, 9:30 A.M.

Dean L. E. Seeley, second vice president, called the third session to order at 9:30 a.m., and introduced E. R. Queer, professor of Engineering Research, The Pennsylvania State College, State College, Pa., who presented his paper, Permeance Measurement Improved by Special Cell, with F. A. Joy, also from State College, as co-author (see Chapter 1371).

Craig L. Taylor, associate professor, Engineering Department, University of California, Los Angeles, Calif., presented the paper, Human Tolerance Limits for Extreme Heat, which he prepared with W. V. Blockley (see Chapter 1372).

N. R. Gay, associate professor of Heat-Power Engineering, Cornell University, Ithaca, N. Y., presented the paper which he and C. O. Mackey prepared, Heat Gains are Not Cooling Loads (see Chapter 1373).

The final paper, Solar Energy Transmittance of Eight-Inch Hollow Glass Block, by G. V. Parmelee and W. W. Aubele (see Chapter 1374), was sum-

marized by Mr. Parmelee, research fellow, A.S.H.V.E. Research Laboratory, Cleveland, Ohio.

Dean Seeley then turned the meeting over to President Stacey, who announced the final registration for the meeting as follows: Members 242; ladies and children 120; guests 42; a total of 404.

RESOLUTIONS

WHEREAS THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has just concluded its 1949 Semi-Annual Meeting in Minneapolis; and

WHEREAS it is fitting and proper that the membership of this Society express its appreciation to all of those organizations and individuals who have contributed to make this meeting an outstanding success in every way; and

WHEREAS the Minnesota Chapter, through its President, Richard C. Jordan, and through its other officers and its members, and especially through its Committee on Arrangements headed by John E. Haines as General Chairman, have so well organized and executed their responsibilities as hosts; and

WHEREAS this meeting has been made significant by the scientific contributions of the authors and discussers at the technical sessions; and

WHEREAS the Chapter ladies have so graciously and charmingly made this sojourn one long to be remembered by the visiting members and their families; and

WHEREAS this entire meeting has been suffused with the glow of hospitality and fellowship, and has been invigorated by the salubrious climate of this most pleasant and genial setting, including, with the charm of a delightful countryside, the amenities and cultural refinements of a great metropolitan center; and

WHEREAS the comfort and well-being of our visiting members and their families have been greatly enhanced through the capable efforts of the hotel managements and staffs; and

WHEREAS all of the visitors have enjoyed and profited by the cultural advantages of this setting:

NOW, THEREFORE, BE IT RESOLVED THAT THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS hereby expresses its sincere appreciation to the Minnesota Chapter, to the authors of papers, to the technical committees, to the Convention Bureau of Minneapolis, to Station KSTP, to the local plants opened for inspection; to the newspapers and trade press for their attendance and splendid coverage of our sessions; and to all of the organizations and individuals enumerated; and

THAT this resolution be spread upon the minutes of this meeting, and the secretary of the Society be instructed to transmit a copy thereof to the president of the Minnesota Chapter.

Respectfully submitted,

RESOLUTIONS COMMITTEE, CARL F. KAYAN, *Chairman*
THOMAS D. STAFFORD, JOHN HAYES CARTER

President Stacey declared that the 1949 Semi-Annual Meeting was adjourned.

HOSPITALITY KEYNOTE

Informal pastimes and good fellowship were the entertainment highlights at the Semi-Annual Meeting in Minneapolis.

The Smorgasbord dinner on Monday night was served to 344 persons present and was conducted in the Swedish tradition.

BANQUET

The Wednesday night banquet on June 22 had no scheduled speakers, although there was a speakers' table, presided over by Bob DeHaven, CBS broadcaster in Minneapolis. He conducted a make-believe radio program with audience participation.

Following the awarding of three golf trophies, the Research Cup, the Eichberg Memorial Cup, and the special Paul Bunyan Cup, a *speaker* then arrived and insisted on being heard. The surprise *speaker* was revealed as humorist Hal Olson, whose jokes and anecdotes provided a pleasant finale to the meeting.

PROGRAM SEMI-ANNUAL MEETING

Nicollet Hotel, Minneapolis, Minn.—June 20-22, 1949

Sunday—June 19

10:00 a.m. Committee on Research—*Hennepin Room*
10:00 a.m. Special Committee on Membership Grades—*Parlor F*
10:30 a.m. Executive Committee—*Parlor C*
1:00 p.m. REGISTRATION—*Mezzanine*
1:00 p.m. Minnesota Chapter Hospitality Room—*Parlor JK* (open until midnight)
1:30 p.m. Council—*Parlor D*

Monday—June 20

8:00 a.m. Minnesota Chapter Hospitality Room—*Parlor JK*
9:00 a.m. REGISTRATION—*Mezzanine*
10:00 a.m. TECHNICAL SESSION—*Grand Ballroom*
Presiding Officer—Alfred E. Stacey, Jr.
Greetings by John E. Haines—Response by President Stacey
Amendment to By-Laws
Determining Performance of a Blower Used with a Furnace, by N. A. Buckley and R. W. Roose
Air Streams from Perforated Panels, by Alfred Koestel, Philip Hermann and G. L. Tuve
Pressure Loss and Air Flow Characteristics of a Box Plenum, by S. F. Gilman, R. J. Martin, W. R. Hedrick and S. Konzo
12:15 p.m. Ladies Luncheon and Card Party—*Minnesota Terrace Room*
12:30 p.m. Men's Buffet Luncheon
Golf Tournament—Research Cup and Eichberg Memorial Cup Matches, Minneapolis Golf Club
1:00 p.m. Children's Tour
2:00 p.m. Inspection Trips
2:00 p.m. TAC on Industrial Ventilation—*Parlor C*
2:00 p.m. TAC on Sound Control—*Parlor D*
2:00 p.m. TAC on Sensations of Comfort—*Parlor E*
6:00 p.m. Social Hour, Dinner, Smorgasbord, Entertainment and Dancing

Tuesday—June 21

8:00 a.m. Minnesota Chapter Hospitality Room—*Parlor JK*
9:00 a.m. REGISTRATION—*Mezzanine*
9:30 a.m. TECHNICAL SESSION—*Grand Ballroom*
Presiding Officer—Lester T. Avery
Report of Committee on Research by L. N. Hunter, Chairman
Earth as a Heat Source or Storage Medium for the Heat Pump, by E. W. Guernsey, P. L. Betz and N. H. Skau
Heat Pump Results in Equitable Building, by J. Donald Kroeker, R. C. Chewning and C. E. Graham
Ground Temperatures as Affected by Weather Conditions, by A. B. Algren
10:00 a.m. }
11:00 a.m. } Ladies Inspection of Betty Crocker's Kitchen
1:00 p.m. Golf (informal)
1:30 p.m. Automobile Tour of the City Lakes and Lake Minnetonka
1:30 p.m. Nominating Committee—*Parlor F*
1:30 p.m. Guide Publication Committee—*Parlor L*
2:00 p.m. TAC on Air Distribution—*Parlor C*

2:00 p.m. TAC on Heat Pump—*Parlor D*
6:30 p.m. Dinner—Reservations at selected eating places
6:30 p.m. TAC on Panel Heating and Cooling—Dinner Meeting—*Parlor F*
7:00 p.m. TAC on Cooling Load—*Parlor C*
7:00 p.m. Launch Trips—Lakes Calhoun, Isles and Cedar
8:00 p.m. Informal Party—*Junior Ballroom, East Room,* and *Parlor JK*
8:00 p.m. Chapter Delegates Committee—*Hennepin Room*

Wednesday—June 22

8:00 a.m. Minnesota Chapter Hospitality Room—*Parlor JK*
9:00 a.m. REGISTRATION—*Mezzanine*
9:30 a.m. TECHNICAL SESSION—*Grand Ballroom*
Presiding Officer—Lauren E. Seeley
Permeance Measurement Improved by Special Cell, by F. A. Joy and E. R. Queer
Human Tolerance Limits for Extreme Heat, by W. V. Blockley and Craig L. Taylor
Heat Gains Are Not Cooling Loads, by C. O. Mackey and N. R. Gay
Solar Energy Transmittance of Eight Inch Hollow Glass Block, by G. V. Parmelee and W. W. Aubele
10:30 a.m. Ladies and Children—Radio Station KSTP
1:00 p.m. Children—Games and Swimming—Lake Nokomis
1:00 p.m. Golf (Informal)
2:00 p.m. Inspection Trips
2:00 p.m. TAC Sub Committees on Cooling Load and Heat Flow Through Glass
3:00 p.m. Ladies Style Show and Tea—*Dayton's Sky Room*
5:00 p.m. Social Hour—*East Room*
7:00 p.m. BANQUET—*Grand Ballroom*
Entertainment and Dancing
Bob DeHaven, radio comedian, master of ceremonies
Presentation of Research and Eichberg Golf Trophies

COMMITTEE ON ARRANGEMENTS

J. E. HAINES, *General Chairman*

L. C. GROSS, *Vice Chairman*

F. B. ROWLEY, *Honorary Chairman*

Banquet—D. B. Anderson, *Chairman;* R. E. Backstrom, G. S. Bean, J. A. Craig, J. C. Davidson, William McNamara, C. P. Petersen, William Sturm.

Entertainment—J. S. Locke, *Chairman;* M. R. Eastin, S. L. Furber, Mrs. J. E. Haines, N. B. Herman, H. K. Knoblauch, L. J. Krause, J. A. Nicols.

Finance—C. T. Lawrence, *Chairman;* C. L. Bensen, Guy M. Brown, C. O. Carlson, D. M. Forfar, H. K. Knoblauch, H. P. Roberts, F. H. Schernbeck.

Ladies—H. K. Johnson, *Chairman;* A. L. Cherne, Mrs. H. E. Gerrish, Mrs. H. K. Johnson, Mrs. J. S. Locke, Mr. and Mrs. A. W. Schultz, H. G. Sierk, H. D. Woolman.

Publicity—H. C. Mills, *Chairman;* J. R. Bergan, R. T. Haley, H. B. Williams.

Reception—A. B. Algren, *Chairman;* R. E. Backstrom, M. H. Bjerken, M. R. Eastin, H. E. Gerrish, W. H. Kliever, F. F. Lange, J. F. Stafford, W. F. Uhl, M. S. Wunderlich, R. T. Squier.

Sports—B. F. McLouth, *Chairman;* E. F. Bell, J. A. Craig, R. T. Haley, V. E. Nelson.

Transportation—G. M. Kendrick, *Chairman;* R. D. Albyn, J. K. Gerrish, J. G. Hamm, H. D. Woolman.

1365

No. 1365

DETERMINING PERFORMANCE OF A BLOWER USED WITH A FURNACE†

By N. A. BUCKLEY* AND R. W. ROOSE,** URBANA, ILL.

IN A TYPICAL forced-warm-air heating system, air is drawn through the return-air ducts and into the furnace through air filters, and is then forced over the heat exchanger surfaces and through the warm air duct system into the space to be heated as shown in Fig. 1. The force required to move the air through this system of ducts and restricted passages is developed by a furnace blower installed inside the casing of the furnace, or in a blower cabinet in close proximity to the furnace.

The resistances placed on the blower by the duct system, external to the blower-furnace combination, are usually predetermined at some arbitrary value by the designer of the duct system. However, the internal resistances of the furnace and blower cabinet can not be predetermined or calculated in advance and must be evaluated by actual tests.

Object and Scope of Investigation. The object of this investigation was to specify a test arrangement and method which would enable blower and furnace manufacturers to determine the performance characteristics of a blower‡ when used in conjunction with a forced-air furnace.

Methods of Determining the Performance of a Furnace Blower. At the time that this project was originated no convenient test method had been formulated to determine the performance of a blower when used in conjunction with a furnace. However, several methods are available for obtaining the performance of a furnace blower alone. Among these, the two most commonly used and accepted are the methods designated as Plates I and V by the *National Associa-*

† This investigation is a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the *University of Illinois* and the *National Warm Air Heating and Air Conditioning Association.*

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** Research Associate in Mechanical Engineering, University of Illinois. Junior Member of A.S.H.V.E.

‡ The term *blower* as used herein refers to a centrifugal fan. The warm-air heating industry uses the term blower to distinguish centrifugal fans, such as are now used in forced air systems, from propeller fans which were originally used.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Minneapolis, Minn., June, 1949.

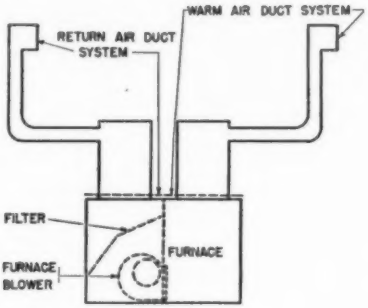

FIG. 1. SCHEMATIC DIAGRAM OF A FORCED AIR HEATING SYSTEM

tion of Fan Manufacturers as shown in Fig. 2, hereafter abbreviated to *N.A.F.M.* Plates I and V methods.[1]

The *N.A.F.M.* recommends that the Plate V method be used in testing and rating furnace blowers, but because of the extensive space and equipment necessary for using the Plate V method, few manufacturers have adopted it. At the present time, furnace blowers are usually tested and rated by the Plate I method. It is recognized that the performance of a furnace blower as obtained by the Plate I method may deviate somewhat from that obtained by the Plate V method, and furthermore, may not be truly representative of the performance of a blower used with a furnace.

The performance obtained by the Plate I method is dependent to some extent upon the area of the blower discharge and the conversion of velocity pressure to static pressure in the test duct, whereas in an actual installation, a considerable portion of the velocity pressure may not be regained.

[1] Exponent numerals refer to References.

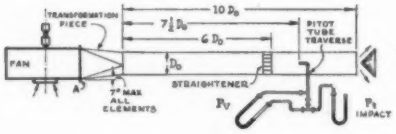

PLATE I

Blowers—Single or Double Inlet

Duct: Total pressure, p_t, by impact tube—Velocity pressure, p_v; Differential reading between impact tube and static tube.

Fan Outlet: P_v; Correct p_v to P_v, the velocity pressure corresponding to fan outlet area, A—P_t; Add to p_t the calculated duct friction and straightener loss—P_s; Subtract P_v from P_t—Correct all pressures to standard air conditions. All additions and subtractions mentioned above are arithmetical. No inlet pressure readings are to be taken and no alteration of designed inlet is to be made.

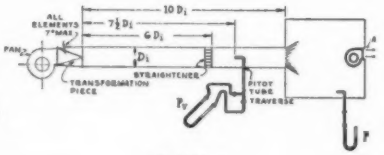

PLATE V

Blowers and Exhausters without Duct

Supply Duct: Velocity pressure, p_v; Differential reading between impact tube and static tube.

Chamber: p by impact tube.

Fan Outlet: P_v; Correct p_v to P_v, the velocity pressure corresponding to fan outlet area, A—P_t; Add p to P_v—P_s; Subtract P_v from P_t—Correct all pressures to standard air conditions. All additions and subtractions mentioned above are arithmetical. Refer to Plate XII for chamber proportions.

FIG. 2. TWO METHODS OF TESTING AND RATING FURNACE BLOWERS

However, since the Plate I method is predominantly used at the present time in rating furnace blowers, it was used in the tests reported in this paper to illustrate the procedure for determining the pressure drop through the blower-furnace combination. The procedure given is applicable to either the Plate I or the Plate V method, and any later adoption of the Plate V method by the furnace blower industry will not invalidate the conclusions of the paper.

The performance of the blower, regardless of the method used to obtain it, can be represented by a curve in which the air quantity in cubic feet per minute is plotted against static pressure. For any given speed, the blower will deliver an amount of air dependent upon the resistance opposed to flow.

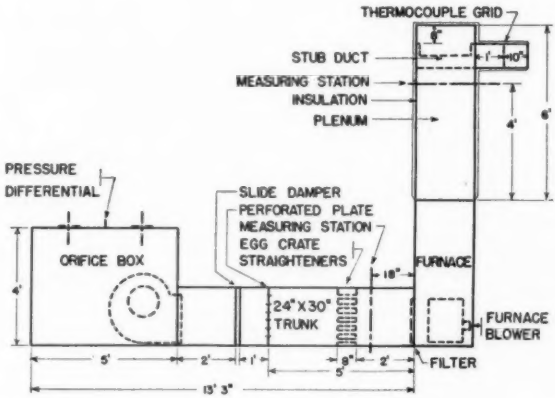

FIG. 3. SCHEMATIC DIAGRAM OF THE HI-BOY FURNACE TEST ARRANGEMENT

Variables Involved in Determining the Performance of a Blower-Furnace Combination. The combination of a blower and furnace can be considered as an equivalent blower unit. As in the case of an isolated blower the performance of the combination can be represented, for a particular blower speed, by plotting the algebraic difference between inlet and discharge pressure obtained by piezometer connection against air quantity flowing. This blower speed corresponds to that required to obtain the performance recommended[2, 3] by the *National Warm Air Heating and Air Conditioning Association* in *Manuals 6* and *7*.

These operating conditions can be summarized as follows: with rated fuel input to the furnace, the combination must be able to provide a static pressure of 0.20 in. of water external to the unit and maintain a temperature rise of 100 deg through the unit from inlet to discharge. When the combination is operated with these limiting conditions, a specific blower speed can be established. The performance of the combination for this given blower speed can be determined by varying the static pressure external to the unit and finding the corresponding quantity of air.

DESCRIPTION OF TEST APPARATUS

Hi-Boy[△] *Furnace Test Arrangement.* The test arrangement for the hi-boy furnace consisted of an orifice box, a trunk duct, the furnace, and a furnace plenum together with stub duct. A schematic diagram of the test arrangement is shown in Fig. 3. The orifice box, 5 ft wide by 5 ft deep by 4 ft high, was provided with: an auxiliary blower and a ¾ hp driving motor inside the box; four orifices, 5, 7, 8, and 11 in. in diameter on top; and a piezometer connection at the center of the top plate.

The trunk duct attached to the orifice box at the auxiliary blower discharge was 24 in. × 30 in. and contained a slide damper, a perforated plate, and egg-

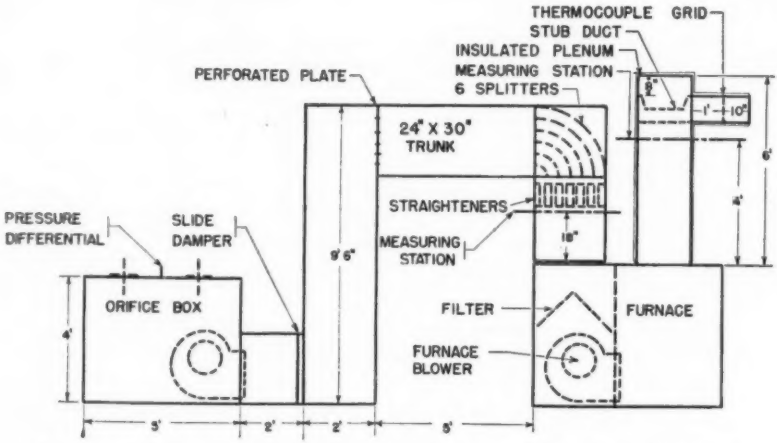

FIG. 4. SCHEMATIC DIAGRAM OF THE LO-BOY FURNACE TEST ARRANGEMENT

crate straighteners. A piezometer ring was installed at the measuring station for determining the static pressure in the trunk duct.

A plenum 6 ft high was installed at the furnace discharge and was covered with two layers of ¼ in. corrugated asbestos insulation, to minimize heat loss from the plenum between the furnace discharge and the temperature measuring station.

Lo-Boy[°] *Furnace Test Arrangement.* The test arrangement for the lo-boy furnace consisting of the orifice box, trunk duct, the lo-boy furnace, the furnace plenum, and stub duct is shown schematically in Fig. 4.

The trunk duct in this arrangement formed an arch to allow entry of the air at the top of the fan compartment. The plenum height was the same as that shown in Fig. 3 but the areas of the plenum and the stub duct were larger to correspond with greater furnace capacity.

[△] Designation of a furnace unit having a blower located below the heating element.
[°] Designation of furnace with blower located at side of heating element.

Instrumentation. The differential pressures of the orifice box, the static pressures in the trunk duct, and the static pressures in the plenum were measured by means of inclined draft gages. Two gages with scale markings from 0 to 0.5 in. of water and two gages with scale markings from 0 to 1.0 in. of water were used. These draft gages were each calibrated with a sensitive micromanometer and correction curves were established. The speed of the furnace blower was observed by means of a speed indicator, reading directly in revolutions per minute.

Thermocouples located at the inlet and discharge of the furnace were used to measure air temperatures. Wet and dry bulb temperatures of the inlet air were obtained by means of a sling psychrometer. A gas meter and stop watch were used to determine the actual gas input volume delivered to the furnace, in cubic feet per hour. At the inlet to the gas meter gas temperature was read with a thermometer and gas pressure was measured by a vertical draft gage. Barometric pressures, used for correcting data to standard conditions, were obtained by means of a mercury barometer.

DESCRIPTION OF EQUIPMENT TESTED

The Hi-Boy Furnace. The hi-boy furnace was tested with three different blowers, referred to as Blowers A, B, and C. The rated capacity of the furnace was 58,000 Btu per hour output with an input of 72,500 Btu per hour. The fuel used was natural gas which was burned in a single port, blue flame tubular burner. As is conventional for hi-boy units, the blower was mounted below the heat exchanger, and the air was discharged upward against the bottom of the fire box.

The heat exchanger was of welded steel construction with four tubes slanting diagonally through it from the center of the front and discharging at the top near the rear of the furnace. The clearances between the radiation jacket and heat exchanger were approximately 1 in. At the discharge of the furnace, the radiation jacket was bent inward to cause the air to pass over a greater amount of heat exchanger surface and to place resistance to the flow of air around the heat exchanger, thereby causing more air to pass through the four heat exchanger tubes.

Two return-air inlets were located on opposite sides at the base of the furnace. One or both inlets could be used in operating the unit although for the present tests only one inlet was used. The filter was installed in the return-air inlet.

Blowers A, B, and C. The three 9-in. blowers tested with the hi-boy furnace were centrifugal upward discharge blowers having forward curved blades. Blowers A and B were the blowers normally installed in the furnace when sold commercially. Blower C, although of the same general type, was specially built for this investigation.

The Lo-Boy Furnace and Blower D. The rated capacity of the lo-boy furnace was 132,000 Btu per hour output with an input of 165,000 Btu per hour. The fuel used was natural gas which was burned in a ribbon-type burner.

A blower of the bottom horizontal discharge arrangement was installed in a cabinet integral to the furnace. The blower discharged into a chamber below the heat exchanger, and against the back of the fire box. The air then passed through the flat-tube heat exchanger and out through the top of the furnace. Filters of the throwaway type were located above the blower, and the return-air inlet was above the filters.

DEVELOPMENT OF PROCEDURE

Effect of Placement of External Resistance and Temperature Rise on Performance. A large number of preliminary tests were conducted to determine the effect on performance of changes in the adjustment of various parts of the test arrangement. Four series of tests were conducted to determine the effect of separately applying the external resistance on the inlet and discharge sides

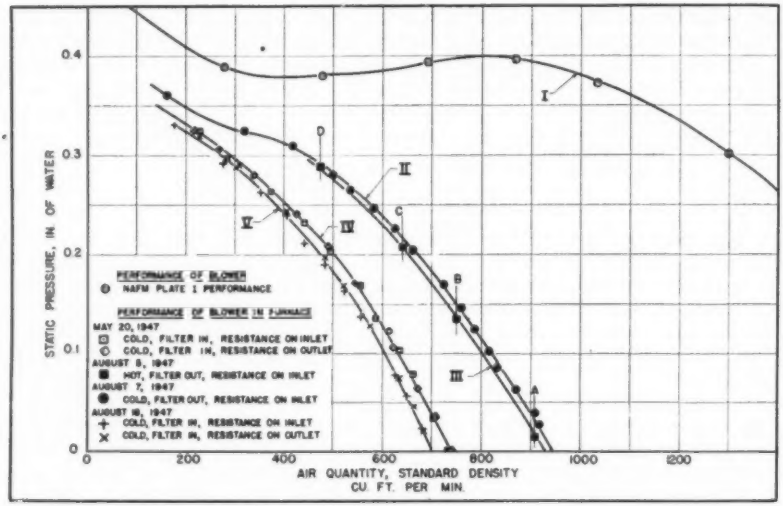

FIG. 5. PERFORMANCE SHOWING EFFECT OF PLACEMENT OF RESISTANCE AND ADDITION OF HEAT

of the combination, and running tests with the furnace both hot and cold. These preliminary tests consisted of the following series:

1. With no heat input and with external static resistance applied at the furnace inlet.

2. With no heat input and with external static resistance applied at the furnace discharge.

3. With rated heat input and with external static resistance applied at the furnace inlet.

4. With rated heat input and with external static resistance applied at the furnace discharge.

The hi-boy furnace was tested in combination with Blower A and the data were plotted as shown in Fig. 5. These tests were conducted both with the filters installed in the unit and with the filters removed.

On May 20, 1947, two series of tests were conducted, one with resistance

on the inlet and the other with resistance on the discharge. The points representing these two series of tests could be plotted on a single curve as shown by Curve IV in Fig. 5. Apparently, the placement of resistance on both the inlet and discharge of the blower-furnace combination gave the same performance.

On August 18, 1947, two series of tests were conducted under similar conditions and these data are represented by Curve V in Fig. 5. Again, the placement of resistance at both the inlet and discharge gave the same performance; however, in this case the performance curves were lower than those for the tests conducted three months earlier. This difference was attributed to the filling of the filter with dust and lint, resulting in an increase in pressure drop through the filter.

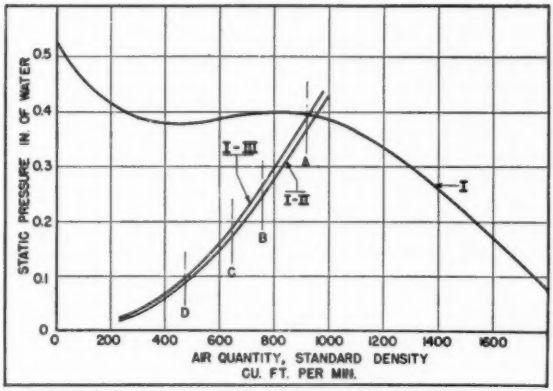

FIG. 6. PRESSURE DROP CURVES SHOWING EFFECT OF ADDITION OF HEAT

Steel plates provided with a large number of ¾ in. holes were therefore substituted for filters. This arrangement eliminated any change in pressure drop such as would be caused by accumulation of dirt in a filter.

The effect of adding heat to the unit is shown by the two performance Curves II and III in Fig. 5 which were established with the filters removed and with rated fuel input to the furnace. The addition of heat to the unit resulted in a decrease of static pressure available external to the unit for the same blower speed and quantity of air handled. This decrease in performance is illustrated by the difference between the pressure drop curves in Fig. 6. Assuming the Plate I performance to be a true indication of the performance of a furnace blower when operated separately, the difference between the Plate I performance curve and that for the blower-furnace combination can be considered as the pressure drop through the blower-furnace combination. Pressure drop Curve (I-II) of Fig. 6 was higher than Curve (I-III) which indicates that the pressure drop through the blower-furnace combination was increased slightly when heat was added.

These results were predicted by S. H. Downs[4] who stated that in a system

so arranged that heat is added or subtracted, heat change is coincidental with the resistance, or

$$r = K_1 \cdot \frac{T_m}{T_1}$$

where

$$T_m = \frac{T_2 - T_1}{\log_e \frac{T_2}{T_1}}$$

or

$$r = K_2 \cdot \sqrt{\frac{T_2}{T_1}} \text{ (approx.)}$$

where

r = system static resistance, inches of water
K_1 = a constant
K_2 = a constant
T_1 = temperature entering the system, degree Rankine
T_2 = temperature leaving the system, degree Rankine
T_m = log mean temperature, degree Rankine

It was also stated by Downs in another paper[5], that a determination of resistance as affected by temperature changes can be made by the proportionalities:

$$\frac{R_H}{R_C} = \frac{T_m}{T_1}$$

or

$$\frac{R_H}{R_C} = \sqrt{\frac{T_2}{T_1}} \text{ (approx.)}$$

where

R_C = static resistance of system when no heat is added, inches of water
R_H = static resistance of system when heat is added, inches of water

Calculations of the temperature ratios $\frac{T_m}{T_1}$ and $\sqrt{\frac{T_2}{T_1}}$ were made at the four points indicated in Figs. 5 and 6 as A, B, C, and D, and the results were tabulated in columns 10 and 11 of Table 1. These calculated values were within 1

TABLE 1. RELATIONSHIP BETWEEN ABSOLUTE TEMPERATURE AND PRESSURE DROP THROUGH A FURNACE

(1) Conditions	(2) cfm	(3) T_1	(4) T_2	(5) T_2-T_1	(6) T_m	(7) R_H	(8) R_C	(9) $\frac{R_H}{R_C}$	(10) $\frac{T_m}{T_1}$	(11) $\sqrt{\frac{T_2}{T_1}}$
A	910	549.5	607.6	58.1	581	0.378	0.359	1.053	1.057	1.053
B	750	548.5	616.0	67.5	587	0.262	0.246	1.064	1.069	1.063
C	640	547.5	625.4	77.9	589	0.183	0.171	1.070	1.075	1.070
D	475	546.1	648.1	101.4	596	0.093	0.081	1.082	1.089	1.081

percent of the ratio $\frac{R_H}{R_C}$ obtained from test data, which is within the accuracy of the instruments used.

The excellent agreement between the actual resistance ratio and calculated temperature ratios indicated that, at least for the given furnace, the procedure for determining the resistance of a furnace when heat is added could be simplified. Some question arose as to the temperatures that should be assumed for the correction ratio. In this connection it should be recognized that the range of possible temperature rises through a furnace is limited to a maximum range of 70 to 100 F. Hence, as shown in Column 10 of Table 1, for test points

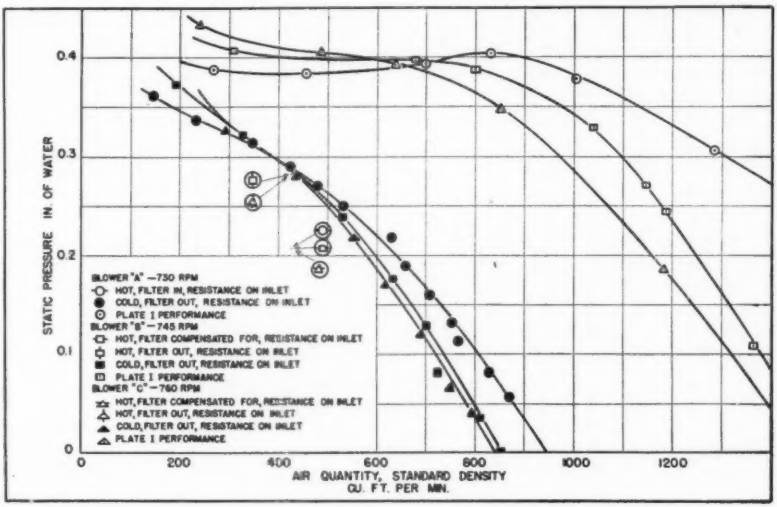

FIG. 7. PERFORMANCE CURVES FOR BLOWERS A, B, AND C

C and D which have temperature rise values of 77.9 and 101.4 respectively, the correction ratio of T_m/T_1 is limited to ratios between about 1.07 and 1.09. For practical purposes, therefore, an average value of 1.08 for this temperature correction ratio was selected and used in subsequent corrections.

Comparison of Performance of Blowers A, B, and C Operated in Combination With the Hi-Boy Furnace. Three blowers were tested in combination with the hi-boy furnace. First, performance curves were established by determining a definite blower speed. This was accomplished by operating the furnace with a 100 deg temperature rise at rated fuel input and with 0.20 in. of water static resistance applied externally to the blower-furnace combination. The unit was then operated with no heat input. By varying the static resistance external to the combination, a performance curve for the combination was established. Finally, the blower was removed from the furnace and an *N.A.F.M.* Plate I performance curve was determined. All data were corrected to standard condi-

tions and to the speed (rpm) determined at rated conditions. These data are shown in Fig. 7.

Comparison of data from the tests with the three blowers indicated that rated operation occurred at the same point for all three blower-furnace combinations. The performances of the blowers when used in combination with the furnace were approximately the same over the range of air quantities obtained for a 70 to 100 deg temperature rise.

Although the performances of the furnace-blower combinations were approximately the same, the Plate I performance curves for these blowers were quite different. The pressure drop through the furnace, obtained by the difference

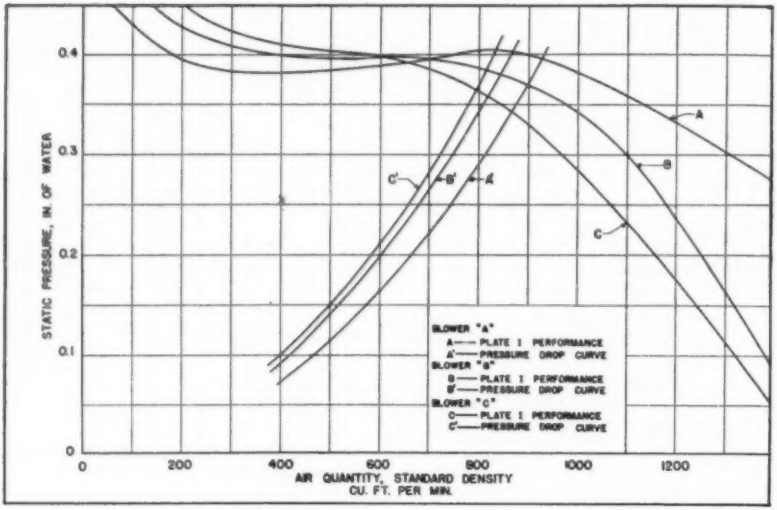

FIG. 8. PRESSURE DROP CURVES FOR BLOWERS A, B, AND C

between the Plate I performance curve and the corresponding performance curve of the blower-furnace combination, was also plotted (Fig. 8).

The difference in pressure drop curves for the three blowers cannot be attributed to any change in the system. Therefore, it may be concluded that blowers of the same type and nominal size, but produced by different manufacturers, do not necessarily perform identically when installed in the same furnace. In another series of tests conducted with a lo-boy furnace, curves similar to those shown in Fig. 8 were obtained.

CONCLUSIONS

1. A test arrangement and method have been developed, which provide for rapid determination of the performance of a blower-furnace combination, and which give an accuracy well within the requirements of industrial applications.

2. The internal resistance of a furnace cannot be calculated. Therefore, the *N.A.F.M.* Plate I or Plate V performance of a blower cannot be reduced by the amount of internal resistance of a furnace in order to determine the performance of a blower-furnace combination.

3. Blowers of the same type and nominal size, but produced by different manufacturers, do not necessarily perform identically when operated with the same furnace.

4. It is advisable to use a substitute for the furnace filters when conducting furnace performance tests. A set of filter substitute plates have been developed for this purpose.

5. Variable resistances can be applied on either the inlet or the discharge of the furnace without changing the performance characteristics of the furnace-blower combination.

6. Increase of resistance due to heating of the air by the furnace can be calculated. Over the narrow range of temperature rises encountered in furnace testing the pressure drop through a blower-furnace combination will be 8 percent greater when operated with heat input than when operated with no heat added. Pressure drop with heat added = (Pressure drop with no heat added) $\times$ 1.08. Therefore, except for establishing rated operating conditions, conducting tests with heat added is not considered essential.

RECOMMENDED PROCEDURE

Establishing the Rated Point of Operation. Based upon the experience gained in conducting the tests on both the hi-boy and lo-boy blower-furnace combinations, a general procedure has been developed. The blower-furnace combination to be tested is installed in the arrangement shown in either Fig. 3 or 4, depending on the type of furnace to be tested. Since the method is the same for both arrangements, the procedure will be outlined for the hi-boy arrangement alone. The seven measuring stations designated in Fig. 9 will be referred to by number in the steps outlining the procedure. The rated point of operation is determined by varying the blower speed until a 100 deg temperature rise is obtained with rated fuel input and a static resistance of 0.20 in. of water applied externally to the unit. The procedure for establishing the rated point of operation may be outlined as follows:

1. Supply the rated fuel input to the furnace as designated by *1* in Fig. 9.

2. Start both the auxiliary blower in the orifice box and the blower under test. Apply an external static resistance of 0.20 in. of water to the blower-furnace combination by varying the closure of the trunk slide damper. The algebraic difference in static pressures as measured at stations *2* and *3* gives the external static resistance.

3. Determine the temperature rise through the unit by the difference in temperatures at *4* and *5*.

If the temperature rise through the furnace exceeds 100 deg, an insufficient quantity of air is being delivered through the furnace. In order to increase the quantity of air flowing, and thus decrease the temperature rise when 0.20 in. of water static resistance is being applied externally to the combination, the speed of the blower should be increased and steps 2 and 3 should be repeated. However, if the temperature rise through the furnace is too small, the quantity of air being delivered through the furnace is too great. In order to decrease the quantity of air delivered through the furnace, the speed of the blower is decreased; steps 2 and 3 are then repeated to determine the new temperature rise. When a 100 deg temperature rise has been attained, the rated point of operation and rated blower speed have been established, and steps 4 and 5 can then be carried out.

4. Obtain the quantity of air being delivered to the furnace, for rated operation, from the orifice differential at *6*.

5. Obtain the blower speed for rated operation of the blower-furnace combination at *7*.

After correcting the data obtained in steps 1 through 5 to standard conditions, the rated point of operation can be plotted as shown by point B in Fig. 10*a*.

Establishing the Performance Curve. Once the rated point of operation has been established, the ratio between the pulley speeds on the blower and on the blower motor should be held constant, but the fuel supply to the furnace should be shut off. No heat should be added to the furnace for any of the remaining tests.

A performance curve for the blower-furnace combination under test should be determined in the following sequence:

1. Adjust the trunk slide damper to the closed position, thus applying the maximum amount of resistance externally to the blower-furnace combination.

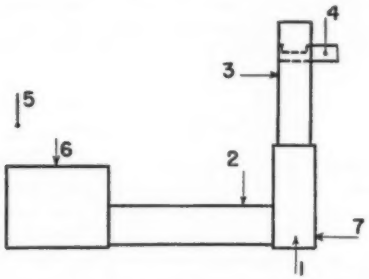

FIG. 9. SCHEMATIC DIAGRAM OF TEST ARRANGEMENT

2. Start the blower which is being tested with the furnace, but not the auxiliary blower in the orifice box. With these operating conditions the following data can be obtained:

a. The quantity of air being delivered to the furnace by means of the orifice differential at *6*.

b. The static resistance applied externally to the blower-furnace combination, by the algebraic difference in static pressure at *2* and *3*, and

c. The speed at which the blower is operating at *7*.

3. Change the closure of the trunk slide damper and repeat step 2, obtaining the quantity of air flowing, static resistances external to the unit, and the blower speed.

4. Having obtained the full range of values available for only the furnace blower in operation, start the auxiliary blower in the orifice box and obtain the full range of values available with both the furnace blower and auxiliary blower operating as outlined in steps 1 through 3.

5. Correct the data obtained in steps 1 through 4 to standard conditions for air and the blower speed established for rated operation. The data when plotted will be similar to the typical performance curve shown on Curve C in Fig. 10*a*.

Pressure Drop through the Unit. The drop in static pressure occurring as the air passes through the blower-furnace combination is determined by the difference between the performance of the blower installed in the furnace, and

the performance of the blower when operated separately. The method of procedure for obtaining the performance of a blower installed in a furnace is outlined in the previous section of this paper, and a representative plot is shown as Curve C in Fig. 10*a*. A representative Plate I performance curve is shown as Curve A in Fig. 10*a*. The difference between the performance of the blower installed in a furnace (Curve C) and the performance of the blower operated separately (Curve A) is shown as Curve D of Fig. 10*b*.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with an investigation of warm-air heating systems being conducted in the Department of Mechanical Engineering at the University of Illinois. This investigation is a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association.*

Acknowledgment is made to Professor S. Konzo for his invaluable assistance in outlining the test objectives and in analyzing the test results. These results will ultimately comprise part of a bulletin of the University of Illinois Engineering Experiment Station.

Acknowledgment is made to the *Furnace Blower Manufacturers Association* for supplying valuable technical assistance and to the various manufacturers who cooperated by furnishing materials and equipment.

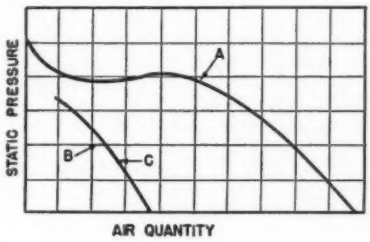

FIG. 10*a*. PERFORMANCE CURVES

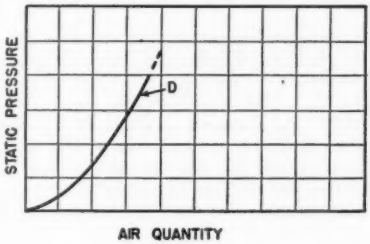

FIG. 10*b*. PRESSURE DROP CURVES

REFERENCES

1. Standard Test Code for Centrifugal and Axial Fans, 3rd Edition. (*National Association of Fan Manufacturers,* Detroit, Mich., 1941.)

2. *Manual 6,* Service Manual for Continuous Air Circulation Technicians, 1st Edition. (*National Warm Air Heating and Air Conditioning Association,* Cleveland 14, Ohio, 1947.)

3. *Manual 7,* Code and Manual for the Design of Warm Air Winter Air Conditioning Systems, 2nd Edition. (*National Warm Air Heating and Air Conditioning Association,* Cleveland 14, Ohio, 1947.)

4. System Characteristics, by S. H. Downs. (Clarage Fan Co., Kalamazoo, Mich., 1933.)

5. Varying Densities Within a System and Effect on Ratio of Opening, by S. H. Downs. (Clarage Fan Co., Kalamazoo, Mich., 1933.)

DISCUSSION

PAUL R. ACHENBACH, Washington, D.C. (WRITTEN): The authors of this paper are to be congratulated for their investigation of a blower applied to a furnace and duct system. Information of this kind has been lacking and there has been misapplication of blowers to furnaces in some cases. The point is not clear that Curves II, III, IV, and V in Figs. 5 and 7 showed the relationship between the air delivery of the blower and the static pressure loss of the duct system external to the furnace. Since the whole substance of the paper is related to the effect of the static pressure loss inside the furnace on the performance of the blower, Figs. 5 and 7 should be labelled in some way to indicate that the curves refer to external static pressure.

D. M. MILLS, Houston, Tex.: I would like to ask for further comment on the relation between actual fan capacities when installed in the furnace, as compared with manufacturers' published ratings.

AUTHOR'S CLOSURE: The complete report of this investigation has been published as Engineering Experiment Station Bulletin 390 entitled "Proposed Arrangement and Method for Determining Pressure Characteristics of a Blower Used with a Furnace" by N. A. Buckley, S. Konzo, and R. W. Roose, by the University of Illinois in June 1950. The questions raised by Mr. Achenbach are covered in detail in this bulletin.

In answer to the comment of Mr. Mills, it may be stated that the objectives of this investigation did not include the comparison of the actual capacities of the blowers determined in this investigation with the published ratings of the blowers by the manufacturers, so this comparison was not made.

1366

No. 1366

AIR STREAMS FROM PERFORATED PANELS

By Alfred Koestel*, Philip Hermann**, and G. L. Tuve†,
Cleveland, Ohio

This paper is the result of research sponsored by The American Society of Heating and Ventilating Engineers in co-operation with the *Case Institute of Technology*.

WHEN a stream of air is discharged from an outlet into still-air space, the velocity of an air particle in the primary jet may be decreased in three ways: (*a*) by dynamic or inertia forces; (*b*) by frictional or viscous forces; and (*c*) by buoyant or gravity forces. If the pressure is constant throughout the stream, the density of the air will vary only with temperature, and hence in an isothermal stream the buoyant forces are eliminated. The Reynolds number affords an index of the relative magnitudes of the two remaining forces, *i.e.*, the dynamic and the frictional forces.

Extensive data on air flow from many types and sizes of outlets have been presented in the previous reports in this series[1,2,3]. These experimental results were successfully correlated on the basis of the theory of conservation of momentum, since the tests represented relatively high outlet velocities and Reynolds numbers.

Air streams discharged from perforated panels of the types sketched in Fig. 1 have been found to behave in a manner similar to the streams from free-open outlets and from grilles, but lower velocities are usually involved. In fact, the perforated panel is an excellent device for producing a large-diameter stream that has a uniform velocity across its entire area. The tiny jets diffuse quickly, (Fig. 2), with a resulting uniform, low velocity equal to Q/A fpm,

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[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Minneapolis, Minn., June 1949.

where Q is the volume in cubic feet per minute discharged through the panel and A is its gross area in square feet.

As the primary air stream progresses away from a discharge outlet, it passes through *three zones* or phases. These three zones have been investigated and described by McElroy[4], and the phenomena were confirmed in the present tests: *First zone,* near the outlet panel, the velocity in the center of the stream remains practically constant, and entrainment or mixing takes place only around

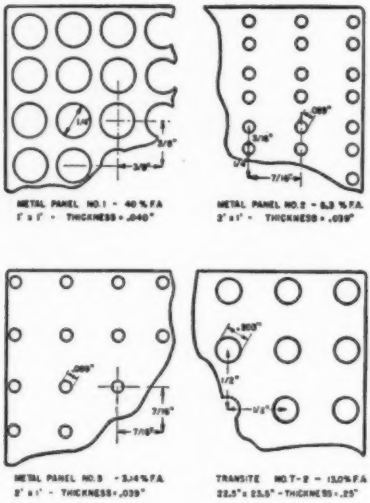
METAL PANEL NO. 1 - 40 % F.A.
1' x 1' - THICKNESS = .040"
METAL PANEL NO. 2 - 8.3 % F.A.
2' x 1' - THICKNESS = .039"
METAL PANEL NO. 3 - 3.14 % F.A.
2' x 1' - THICKNESS = .039"
TRANSITE NO. T-2 - 13.0 % F.A.
22.5" x 23.5" - THICKNESS = .25"

FIG. 1. TYPICAL PERFORATED PANELS TESTED

the stream perimeter; *Second zone,* at a greater distance from the outlet, the maximum or centerline velocity decreases approximately as the square root of the distance. This zone is comparatively short. *Third zone,* the maximum residual velocity varies inversely as the distance X from the outlet, *i.e.,* $V_x/V_o = C/X$. This is the common equation for the throw of an air stream, as given in the HEATING, VENTILATING, AIR CONDITIONING GUIDE*. By the time the third zone is reached, the stream has become very nearly circular, irrespective of the shape of the outlet.

As the residual velocity and, hence, the Reynolds number of the stream in this *third zone* decreases, a point is finally reached at which viscous forces become important. As shown in Fig. 3, the actual velocities then fall below those indicated by the straight-line relationship. Previous reports in this

* Published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 51 Madison Ave., New York 10, N. Y.

series[1-3] have dealt chiefly with the character of air streams in the *third zone,* at velocities above that at which the viscous forces become significant.

In tests of nozzles and grilles, most of the measurements are made at a considerable distance from the panel, because this is the important part of the

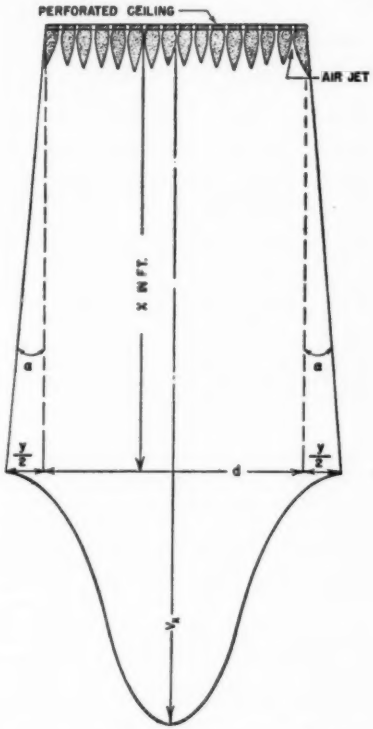

FIG. 2. SKETCH OF DISCHARGE AIR STREAM LEAVING PERFORATED PANEL

stream as far as air mixing, drafts and temperature gradients are concerned. But with perforated panels, which have a larger gross area, the zones close to the panel face are longer and become more important, hence for the panels in Fig. 1 all three of the characteristic zones have been investigated and the results are reported herewith.

THEORY AND ANALYSIS

Fig. 2 is a diagrammatic sketch showing the various air jets issuing from the perforations in a panel, and diffusing or flowing together near the panel

TABLE 1. DESCRIPTION OF PANELS TESTED

PANEL NUMBER	PANEL AREA SQ FT	PERCENTAGE OF FREE AREA	MATERIAL	PANEL THICKNESS IN.	PANEL DIMENSIONS IN.	HOLE SIZE IN.	AVERAGE COEFFICIENT OF DISCHARGE
1	1	40.5	Metal	0.040	12 x 12	0.25	0.83
2	2	6.30	Metal	0.039	24 x 12	0.089	0.75
2A	1	6.30	Metal	0.039	12 x 12	0.089	0.75
4	2	6.22	Metal	0.039	24 x 12	0.125	0.62
5	2	3.14	Metal	0.039	24 x 12	0.089	0.710
T-2	3.67	13.0	Fiberboard (transite)	0.25	22.5 x 23.5	0.203	0.75
T-3	14.65	13.10	Fiberboard (transite)	0.25	46 x 45.875	0.203	0.78
T-4	8.96	12.4	Fiberboard (transite)	0.25	36⅛ x 35¾	0.197	0.79

Test Conditions: For Panel Nos. 1, T-2, T-3 and T-4, the air stream was projected horizontally into open space. For Panel Nos. 2, 2A, 4, and 5, the air stream was projected downward into an enclosed space. All panels were tested under isothermal conditions.

face to form one homogeneous stream. This stream will eventually become circular, although if the panel is rectangular, a distance of several equivalent diameters may be traversed by the stream before it assumes the circular shape. In the following analysis, it is assumed that the diffusion of individual jets takes place into a stream area determined by the area of the panel. This assumption was supported by tests on several sizes and shapes of panels. (See Table 1).

The symbols to be used (see Fig. 3) are as follows:

X = distance from panel face to V_x.

V_x = maximum residual velocity of the discharge air stream at distance X from the panel.

V_o = maximum outlet velocity at the jet contraction of the perforations.

(PA) = gross *panel area* of the perforated panel.

α = angle of expansion of the discharge air stream, assumed to be constant for all panels.

d = equivalent diameter of the panel, $d = \sqrt{\frac{(PA)\,4}{\pi}}$

K_1 = ratio of the average velocity of the discharge air stream at distance X to the maximum velocity V_x. $K_1 = \frac{V \text{ average}}{V_x}$

%(FA) = percentage of *free area* or ratio of the total area of the perforations to the gross panel area. *Note:* Used as a decimal in the calculations.

C_d = coefficient of discharge for the perforated panel.

$y = 2\,X \tan \alpha$.

A_x = circular area of the discharge air stream at distance X.

$$(d + y)^2 \frac{\pi}{4} = A_x$$

Writing the momentum equation in the form, $Q_o\ V_o = Q_x\ V_x$ for a perforated panel, the following analysis may be set forth:

$$V_o \left[V_o \times C_d \times \% \text{ (FA)} \times \text{(PA)} \right] = K_1\ V_x \left[K_1\ V_x\ (d + y)^2\ \frac{\pi}{4} \right]$$

$$V_o^2\ C_d\ \%\ \text{(FA)} \cdot \text{(PA)} = K_1^2\ V_x^2\ \frac{\pi}{4}\ (d + y)^2$$

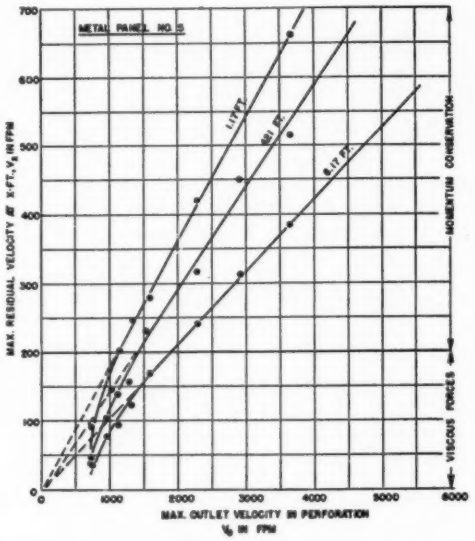

FIG. 3. MAXIMUM RESIDUAL VELOCITY AT A FIXED DISTANCE FROM PANEL, AS DETERMINED BY HEATED THERMOMETER ANEMOMETER

$$V_o \sqrt{C_d\ \%\ \text{(FA)}} \cdot \sqrt{\text{(PA)}} = K_1\ V_x \sqrt{\frac{\pi}{4}}\ (d + y)$$

Substituting for y and d,

$$V_o \sqrt{C_d\ \%\ \text{(FA)}} \cdot \sqrt{\text{(PA)}} = K_1\ V_x \sqrt{\frac{\pi}{4}} \left(\sqrt{\frac{\text{(PA)}4}{\pi}} + 2X \tan \alpha \right)$$

$$V_o \sqrt{C_d\ \%\ \text{(FA)}} = K_1\ V_x \left(1 + 2 \tan \alpha \sqrt{\frac{\pi}{4}}\ \frac{X}{\sqrt{\text{(PA)}}} \right)$$

$$\frac{1}{K_1 \left(1 + 2 \tan \alpha \sqrt{\frac{\pi}{4}}\ \frac{X}{\sqrt{\text{(PA)}}} \right)} = \frac{V_x}{V_o}\ \frac{1}{\sqrt{C_d\ \%\ \text{(FA)}}} \quad \ldots\ldots\ldots\ (1)$$

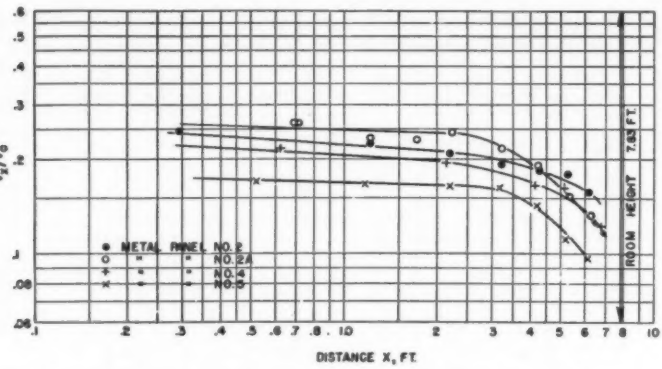

FIG. 4. VARIATION OF CENTER-LINE VELOCITY V_x WITH DISTANCE; (AIR STREAM PROJECTED DOWNWARD FROM AN 8-FT CEILING WITH VELOCITY V_o AT PERFORATIONS)

For the stream at a distance X from a given panel;*

$$\frac{V_x}{V_o} = \frac{\text{a constant}}{X}$$

Two dimensionless parameters are obtained from this derivation, *viz.* $X/\sqrt{\text{panel area (PA)}}$ and $1/\sqrt{C_d \text{ percent free area (FA)}}$, (see Equation 1), and these may then be used in correlating test data. This form of equation corresponds to Equation 25 presented by Rydberg and Norbäck[5].

* HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1949, Chapter 40—Air Distribution, p. 783.

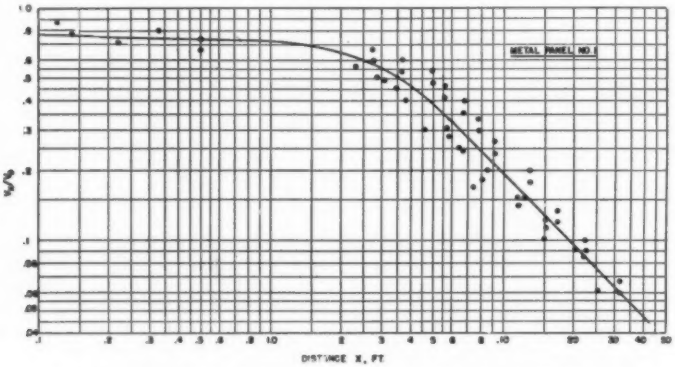

FIG. 5. VARIATION OF CENTER-LINE VELOCITY V_x WITH DISTANCE (AIR STREAM PROJECTED HORIZONTALLY INTO OPEN SPACE WITH VELOCITY V_o AT PERFORATIONS)

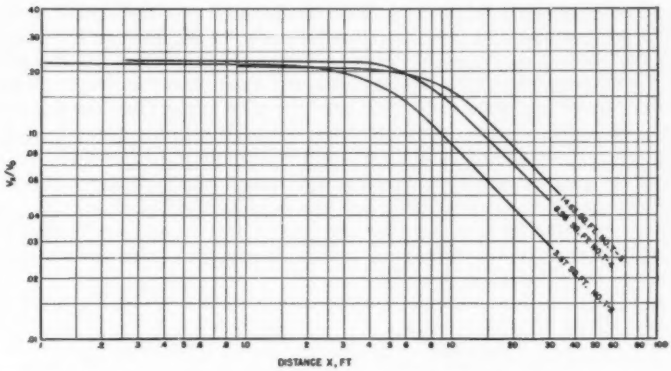

FIG. 6. VARIATION OF CENTER-LINE VELOCITY V_x, WITH DISTANCE, FOR VARIOUS PANEL AREAS

A correlation of the test data on eight panels, based on Equation 1, is given herewith. The dimensions of the test panels are identified in Table 1. Stream velocities below 200 fpm are plotted separately, (Fig. 8), to indicate the magnitude of the viscous effects.

Since the two parameters of Equation 1 are constant for any given panel, it would be expected that a smooth curve should be obtained when the maximum residual velocity at a given cross-section is plotted against the distance from the panel face. This is confirmed by the curves of Figs. 4, 5, and 6.

DATA AND RESULTS

A comparison of stream performance data from four perforated metal panels is presented in Fig. 4. Here the maximum residual velocity is plotted non-

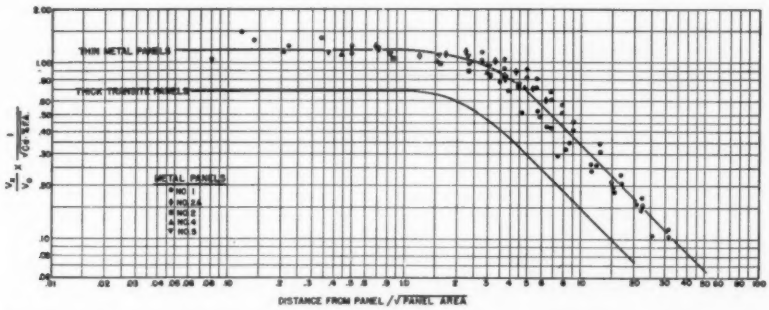

FIG. 7. COMPLETE RESULTS OF TESTS ON 8 PERFORATED PANELS, CORRELATED BY THE USE OF DIMENSIONLESS PARAMETERS

dimensionally against the distance from the panel face. The stream from the small, 1 sq ft panel (No. 2A) clearly attains the third phase or *third zone* in which the velocity decreases in direct proportion to the distance. The streams from the other panels which have an area of 2 sq ft, have barely completed the second or transition phase. In these tests the total throw distance was limited by the low ceiling height. The increased diffusion obtained by using a panel with a low percentage of free area is shown by the curve for Panel No. 5.

When the perforations occupy a large percentage of the gross panel area, the velocity in the *first zone* near the panel face remains high, as shown for a panel of 40 percent free area in Fig. 5. Since these test results were obtained by blowing horizontally into free-open space, the nature of the stream was not limited by floor-to-ceiling distance.

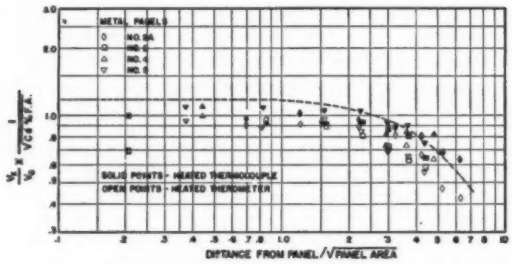

FIG. 8. EFFECT OF VISCOUS FORCES AT LOW VELOCITIES (DOTTED LINE FROM FIG. 7. VELOCITIES BETWEEN 40 AND 200 FPM)

The effect of panel area is indicated by the direct comparisons of Fig. 6. These results were obtained with three transite panels of about 13 percent free area each, but with gross areas in the ratio of 1:2.5:4. The streams from the larger panels are projected a greater distance because of the greater length of each of the three stream zones or phases.

The correlation of test results for all eight perforated panels is shown in Fig. 7. Here the results are made directly comparable by expressing the distance from the panel face in effective diameters, $X/\sqrt{(PA)}$, and by multiplying the velocity ratio V_x/V_o by the diffusion parameter $\frac{1}{\sqrt{C_d\%(FA)}}$ (see Equation 1).

The panel thickness has a marked effect on the diffusion near the panel face, as shown by the lower curve for the relatively thick transite panels. This increase in diffusion explains the position of the horizontal part of the curves in Fig. 6, when these are compared with the curves of Figs. 4 and 5.

As already indicated in Fig. 3, the test points for residual velocities below about 200 fpm deviate from the curves of Figs. 4 to 7. These deviations are shown by Fig. 8, where the dotted curve (from Fig. 7) affords a direct com-

parison. Most of the experimental points fall below this dotted curve, as should be expected when viscous friction decreases the jet velocities. The test results from the heated thermometer and heated thermocouple are shown separately in this plot in order to present a comparison of the two instruments at velocities between 40 and 200 fpm.

Fig. 9 again presents the results of the four metal panels tested within the enclosed space, (see also Fig. 4). The heated thermometer and the heated thermocouple readings are shown separately in Fig. 9, to give another comparison of the two instruments, this time for velocities above 200 fpm.

EXPERIMENTAL METHODS

Velocity traverses of the air stream were taken at several different distances from the face of the panel. Such runs were made at various static pressures

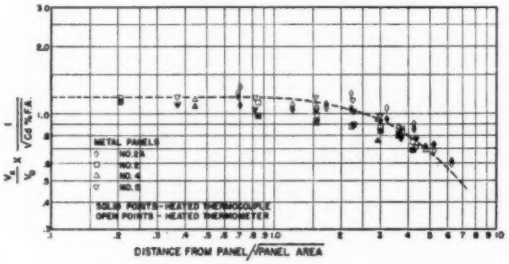

FIG. 9. TESTS ON FOUR METAL PANELS (RESULTS FROM HEATED THERMOMETER AND HEATED THERMOCOUPLE SHOWN SEPARATELY)

using both a heated thermometer (thermo-anemometer) and a heated thermocouple of commercial type. To position the instruments, in the perforated ceiling tests, an 84-in. square grid was suspended from the ceiling in such a manner that it could be set at any desired level from the floor and still be centered upon the perforated panel mounted in the ceiling. The grid was divided into 3-in. squares by fine thread.

The panels were mounted centrally in the ceiling which formed the lower side of a large plenum. The air discharged downward into an enclosed room approximately 18 × 12 × 8 ft, and was exhausted through large outlet grilles in the baseboard at one end of the room.

The procedure was to set the grid level and pressure differential and then take velocity readings with the heated thermocouple positioned in the stream. Readings were taken with the instrument pointing into the moving air stream at the center of each of the grid squares where the velocity was 50 fpm or better. The heated thermometer was used to determine the peak velocity at the location previously indicated by the heated thermocouple.

The calibration of these two instruments was similar to that procedure followed by Seigel[6] on anemometer calibration. The velocity distribution across the test section was very uniform. Air flow was measured by the pressure drop across a calibrated 4-in. orifice mounted downstream of the test section. A micromanometer which checked exactly with a hook gage was used to measure the pressure.

The heated-thermocouple anemometer consisted of series thermocouples mounted in the open-end of a half inch metal tube, with a provision for electrical

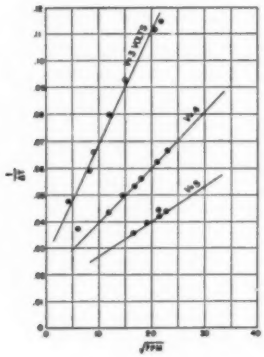

FIG. 10. CALIBRATION OF HEATED THERMOMETER AT VARIOUS IMPRESSED VOLTAGES. (ΔT TEMPERATURE DIFFERENCE BETWEEN HEATED AND UNHEATED THERMOMETER)

heating of the hot junctions due to the electrical resistance of the fine thermocouple wire itself. The magnitude of the d-c thermal voltage generated by the pile is indicated on a calibrated dial reading in feet per minute.

The heated thermometer anemometer was identical to the one described by Yaglou[7]. It consists of two glass thermometers one of which is wound with a manganin wire heating coil about its bulb. A direct current is passed through the coil by impressing a controlled voltage (1.5 to 6 volt) with batteries. The heated thermometer is cooled by the air flow. The temperature difference between the heated thermometer and the ambient thermometer is dependent upon the air velocity. Thus, the higher the velocity the smaller the temperature difference.

The heated thermometer calibration curves (Fig. 10) show close agreement with the curves based on the equation developed by Yaglou[7]. The relationship is linear for all practical purposes.

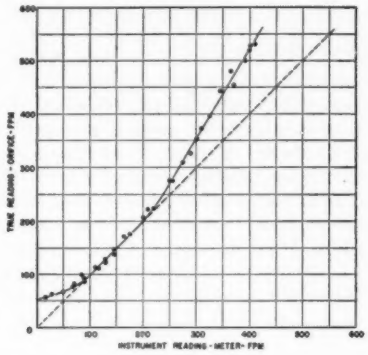

FIG. 11. CALIBRATION OF HEATED THERMOCOUPLE

The calibration curve for the heated thermocouple (Fig. 11) shows very close agreement between the indicated reading and the true reading from 85 to 220 fpm. This calibration was repeated about half way through the tests and the curve was reproduced, indicating that the instrument characteristics were constant with time.

The reading of the heated thermocouple anemometer was subject to the observer's judgment of a time average, under fluctuating conditions. Two observers could differ as much as 50 to 100 fpm for a given reading. This anemometer is more of a directional type instrument because of the nozzle-like shield around the thermocouple mounting. The heated thermometer is less directional and has a much greater mass and, therefore, gives a more completely integrated value. Because of these characteristics, the heated thermometer was preferred for use in determining average peak velocity.

The perforated panels were calibrated by mounting them on a large plenum chamber, metering the air by means of a calibrated orifice, and measuring the

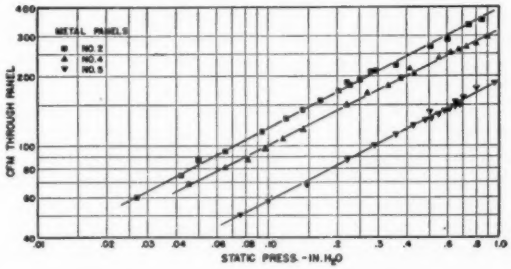

FIG. 12. CALIBRATION OF PERFORATED PANELS ON PLENUM CHAMBER AGAINST STANDARD ORIFICES

static pressures with calibrated inclined manometers. From these values a discharge coefficient was determined at various rates of flow. From Fig. 12 it is evident that the slopes of the curves are very close to 0.5 and indicate a nearly constant coefficient of discharge for a given panel.

In addition to the perforated metal panels, several perforated transite panels were tested. In this case, the air was discharged horizontally into an open space. The panels were mounted on a large plenum. The air flow was metered by means of an orifice, and pressure readings were taken from calibrated inclined manometers. A bridled vane anemometer was used for taking velocity measurements. The anemometer was mounted on a movable stand at centerline height of the perforated board. At fixed distances from the panel, velocity profiles were obtained. The perforated transite panels were calibrated similarly to the perforated metal panels.

DISCUSSION AND CONCLUSIONS

These test data on several types and sizes of perforated panels furnish a confirmation of the analysis used in deriving Equation 1. A method is, therefore, available for estimating the performance of perforated panels, within the range of conditions covered by these tests, *viz.*, 3 percent to 40 percent free area, 1 to 14 sq ft panel size, and up to 3500 fpm velocity through the perforations. Moderate variations in the panel shape or the hole spacing or configuration had no apparent effect on the performance. If tests were made on panels with very small holes, so widely spaced that the individual jets would be prevented from converging into a single stream, the results would no doubt be different, but such panels would usually not be practicable.

It will be noted that when the ratio of velocities V_x/V_o for the thin metal panels was multiplied by the diffusion parameter $1/\sqrt{C_d\%(\mathrm{FA})}$ the result was greater than unity.

For the *first zone* near the panel this product was about 1.18. McElroy termed this ratio, in the *first zone*, the *center constant* and also reported values above unity[4], (between 1.0 and 1.2 depending on the conditions of approach). The length of the *first zone* or phase was somewhat shorter in the present tests than those given by McElroy. The *center constant* for the thick transite panels was found to have a much lower value, approximately 0.72. This was probably due to the occurrence of the jet contraction and the re-expansion within perforations instead of a normal expansion in the discharge space as is the case for the thin metal panels. The fuzziness of the holes may also have been a contributing factor.

In the case where low residual velocities are desired near the panel discharge, for large volumes of primary air supply, the optimum panel design and arrangement would be obtained if panels of small area and low percent free area were spaced in such a way as to prevent the streams of each outlet from converging and forming higher stream velocities. With such small panel areas the phase lengths are decreased, and with low free areas the diffusion is increased.

When curves such as those in Fig. 7 are used for design purposes involving residual velocities below 100 fpm, they will give results on the safe side. On the other hand the curves must be used with care for cooling applications. The velocity ratio $\left(V_x/V_o/\sqrt{C_d\%(\mathrm{FA})}\right)$ is essentially equal to the ratio of tem-

perature gradients, hence if $\left(V_x/V_o/\sqrt{C_d\%(\mathrm{FA})}\right)$ is near unity within the zone of occupancy, only a very small temperature difference is permissible if drafts are to be avoided. However, if low values of $\left(V_x/V_o/\sqrt{C_d\%(\mathrm{FA})}\right)$ are reached, say 0.1 to 0.3, before the stream enters the zone of occupancy, the temperature of the air discharged from the perforations may be correspondingly lowered below the temperature of the room.

Further investigation of the non-isothermal conditions is desirable to establish the design criteria for heated air and cooled air. Further research, of interest from both the theoretical and practical viewpoints, should be conducted on (*a*) stream shapes and profiles; (*b*) effects of hole spacing and aspect ratio; and (*c*) flow in the viscous and transition regions (*second zone*). This further work will give a more firm basis for determining the best size and spacing of perforated panels for year-round air conditioning.

DESIGN EXAMPLE

Given

A room is to be supplied with conditioned air at a rate equal to 1.5 cfm of air per square foot of gross floor area. Perforated ceiling panels will be used, and two styles are available, one with 5 percent free area, the other with 15 percent free area. The residual air velocity at a level 6 ft above the floor must not exceed 75 fpm. Two possible ceiling arrangements are to be considered:

1. Low Ceiling Arrangement—A furred ceiling will be used with a floor-to-ceiling height of 8 ft.
2. High Ceiling Arrangement—The floor-to-ceiling height will be 20 ft.

Required

The selection of panel sizes and air velocities through the perforations. No air-supply panel is to exceed 48 in. in length (or width).

Solution (based on Fig. 7)

V_x in all cases is 75 fpm (The residual velocities determined from Fig. 7 will give conservative values for velocities below 200 fpm).

Assume $C_d = 0.75$ for both panels

The diffusion parameters for the two panels are:

5 Percent (FA) Panel,

$$\frac{1}{\sqrt{C_d \times \%\,(\mathrm{FA})}} = \frac{1}{\sqrt{0.75 \times 0.05}} = 5.15$$

15 Percent (FA) Panel,

$$\frac{1}{\sqrt{C_d \times \%\,(\mathrm{FA})}} = \frac{1}{\sqrt{0.75 \times 0.15}} = 2.98$$

The air quantity flowing from each panel is:

$$\mathrm{cfm} = (\mathrm{PA})\ \%\ (\mathrm{FA})\ C_d\ V_o$$

The static pressure required for each panel is determined from the V_o in the perforations:

$$V_o = \sqrt{2gh}$$

SELECTION

On the basis of the tabulated results, for the conditions given in this example, the most economical selection of the panel type would be based on the percentage of ceiling to be perforated and on the static pressure required for a given quantity of primary air. The required panel area would be an indication of the

TABULATED RESULTS OF COMPUTATIONS

8 FT CEILING X = 2 FT	PANEL SIZE—FEET		
	1 x 1	2 x 2	4 x 4
$\frac{X}{\sqrt{(PA)}}$	2	1.0	0.5
$\frac{V_x}{V_o}\ \frac{1}{\sqrt{Cd\ \%\ (FA)}}$	1.1	1.2	1.2
V_o at 5 percent (FA) fpm	351	322	322
Static pressure required, inches H_2O	0.008	0.006	0.006
Cfm at 5 percent (FA) per panel	13.2	48.4	193.0
Square feet of floor area served per panel	8.8	32.2	129.0
Percent of ceiling to be perforated	11.4	12.4	12.4
V_o at 15 percent (FA) fpm	203	186	186
Static pressure required, inches H_2O	0.003	0.002	0.002
Cfm at 15 percent (FA) per panel	22.8	83.6	335.0
Square feet of floor area served per panel	15.2	55.6	223.0
Percent of ceiling to be perforated	6.6	7.2	7.2

TABULATED RESULTS OF COMPUTATIONS

20 FT CEILING X = 14 FT	PANEL SIZE—FEET		
	1 x 1	2 x 2	4 x 4
$\frac{X}{\sqrt{(PA)}}$	14	7	3.5
$\frac{V_x}{V_o}\ \frac{1}{\sqrt{Cd\ \%\ (FA)}}$	0.25	0.48	0.87
V_o at 5 percent (FA) fpm	1540	804	443
Static pressure required, inches H_2O	0.144	0.040	0.012
Cfm at 5 percent (FA) per panel	57.7	120.5	266.0
Square feet of floor area served per panel	38.5	80.3	177.3
Percent of ceiling to be perforated	2.6	4.98	9.01
V_o at 15 percent (FA) fpm	894	466	257
Static pressure required, inches H_2O	0.048	0.013	0.004
Cfm at 15 percent (FA) per panel	100.0	210	462
Square feet of floor area served per panel	66.9	139.2	308
Percent of ceiling to be perforated	1.5	2.9	5.2

installation costs, and the air quantity and static pressure would determine the cost of operation (supply fan power).

In all cases, the 1 × 1 ft panel is the most economical from the standpoint of panel surface required. The static pressures for the 1 sq ft panels are greater than those of the larger panels, thus indicating higher required fan power, but all pressures are low, so that this factor may be neglected in any economic considerations for this example. The amount of mixing of the primary air

would also be more complete because the mixing lengths are smaller for the 1 × 1 ft panels than for the larger panels.

Other factors beyond the scope of this paper may be entailed in this particular design example. For example, the values of the ratio, $X/\sqrt{(PA)}$, for the 8 ft ceiling are relatively low, and indicate that the residual velocity is in the first phase or mixing zone in which the center velocity remains constant with distance. If the temperature of the discharge air in this case were much below that of the room, the lack of mixing of primary air with relatively warm secondary air may produce conditions of discomfort to any occupants located below the center of the perforated panel.

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7. The Heated Thermometer Anemometer, by C. P. Yaglou (*The Journal of Industrial Hygiene and Toxicology*, October 1938).

DISCUSSION

WILLIAM WALLACE, II, Durham, N.C.: Do the authors expect to extend their study to cover panels of larger area?

E. J. KUREK, Chicago, Ill.: My associates have had considerable success in using perforated panels varying in size from 2 x 4 ft to 3 x 6 ft at perforation velocities of 150 to 200 fpm. Under these conditions, and with temperature differences up to 20 deg, and with 8 ft ceiling height, the air velocity at or below the breathing level had been below 50 fpm, and the temperature gradients in the same zone had been from 1 to 2 deg.

We hope to see further data developed on temperature distribution in non-isothermal streams to correlate analytically with what we have known experimentally.

CHARLES S. LEOPOLD, Philadelphia, Pa.: Distributing air uniformly through an entire perforated ceiling results in low average downward velocity and the delivery of air to the conditioned space at a higher temperature than would result if the air were delivered through a portion of the ceiling only. The air is warmer because a large portion of the internal cooling load is in the form of radiation, both directly from the

source, such as lights, or indirectly from the walls and floor. This radiation tends to warm the ceiling which, in turn, warms the air before it is delivered to the room. Furthermore, if the perimeter losses of the building are met by other means, such as radiators, the air delivered to the interior of the building will be at, or below, room temperature, even in winter.

The full perforated ceiling can be considered as the basis on which we can evaluate the equivalence or compromise in results to be obtained by lesser areas of air introduction.

LESTER T. AVERY, Cleveland, Ohio: Our firm has been using the perforated panel method of distribution for some time, including ducts and sections, as well. It has been found that this is an effective way of handling the cooling load, since it takes advantage of the cooling panel, itself, in addition to providing uniform air distribution.

The paper does not state at what temperature the air was used and I assume that the temperature of air was the same as the temperature of the room in which the tests were conducted. I would like to ask if further tests will be made with warm air or cold air, or whether any tests were made in which the air was ejected horizontally from the sidewall of the duct.

When using the perforated board method of distribution, we have found that it is important to maintain good filtering.

1367

No. 1367

PRESSURE LOSS AND AIR FLOW CHARACTERISTICS OF A BOX PLENUM‡

By S. F. GILMAN,* R. J. MARTIN,** W. R. HEDRICK,† AND S. KONZO,†† URBANA, ILL.

BOX plenums are often used as central air distributing compartments for individual ducts when located in (1) attics, in conjunction with either *overhead supply* or ceiling panel systems; (2) ceilings of halls or closets in basementless homes; and (3) in basements, when the conditioner is not centrally located or the basement height is not sufficient to accommodate a bonnet. The term *box plenum* as used in this paper refers to a type of plenum having a large cross-sectional area relative to the inlet area. The cost of the box plenum system is often substantially less than that of the conventional system, since the box is of rectangular construction without transition sections, and the branch take-off fittings consist of simple butted connections.

The design of such plenums is hampered by the lack of information available to the engineer. Although abrupt expansion and contraction loss data are available and could conceivably apply to the respective entering and leaving losses of a plenum, no data are available for evaluating (1) the turbulent mixing of the air in a plenum, (2) the effect of the number, size, and location of the outlets on the performance of a plenum, and (3) the effect of the physical dimensions of a plenum, such as length, on the pressure loss and air flow characteristics.

In order to extend present knowledge of the performance characteristics of

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‡ This investigation is a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the *University of Illinois* and the *American Gas Association*.

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Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Minneapolis, Minn., June, 1949.

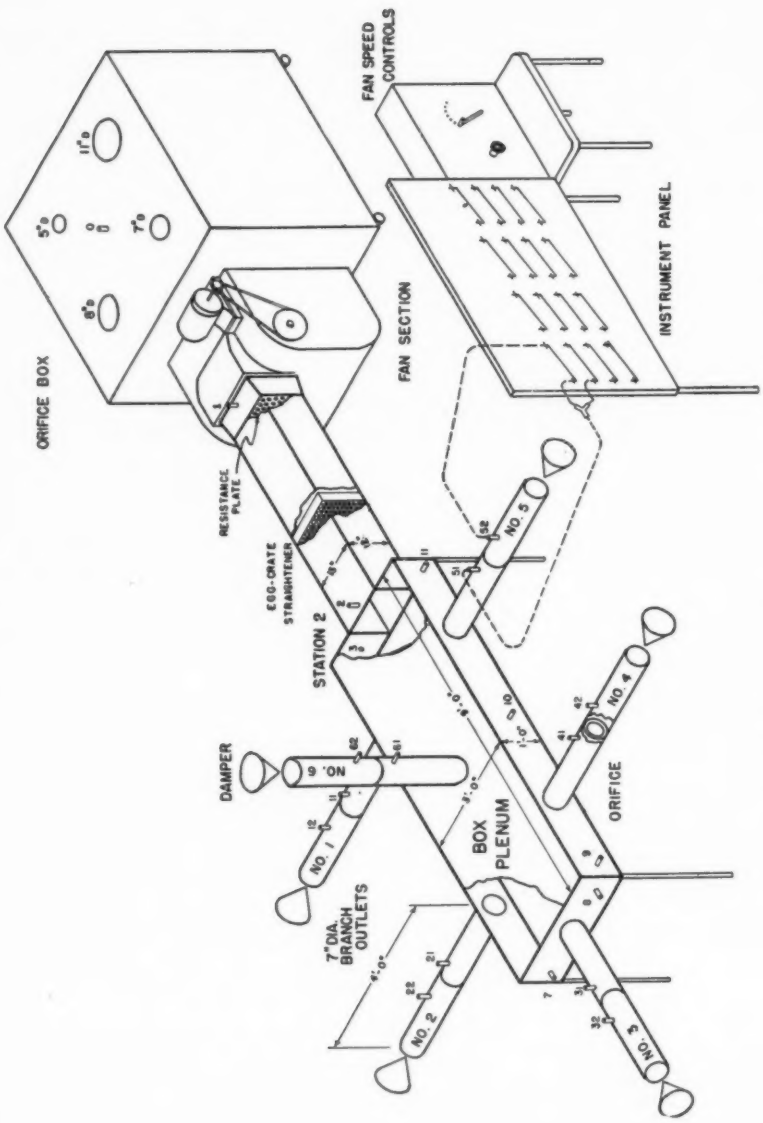

FIG. 1. DIAGRAM OF GENERAL ARRANGEMENT FOR STUDY OF BOX PLENUM

plenums, an investigation of a box plenum with six branch outlets has been conducted. The performance characteristics obtained include the distribution of air to the outlets, the flow pattern inside the plenum, and the evaluation of the pressure losses. The factors investigated and discussed in this paper are:

1. Performance characteristics of a box plenum having the conventional abrupt expansion entrance.

2. Comparative performance of five entrance sections.

3. Performance characteristics of an internal expansion section and tentative design data for its use.

DESCRIPTION OF TEST APPARATUS

The general test arrangement is shown in Figs. 1 and 2. The four orifices in the orifice box were used to measure the total air quantity entering the system, and were used singly or in parallel, depending upon the range of volumes used. The centrifugal blower in the fan section discharged air from the orifice box into the trunk duct leading to the box plenum. Two adjustable rheostats, in conjunction with a d-c motor, provided fan speed variation over a wide range. The resistance plate and egg-crate straightener provided uniform flow conditions at Station 2. The top and bottom of the box plenum were made of transparent material to permit visual observation of directional indicators consisting of 480 silk strings uniformly spaced inside. The six branch outlets were made as exactly alike as possible and were located in a symmetrical pattern about the plenum. The air quantity leaving the system through any outlet was measured by means of an orifice in the outlet. The dampers were adjustable and were used to control the volume flowing in the outlets.

The static pressures in the system were measured by piezometer rings at Stations 1, 2, 61, and 62, and by static pressure taps at all other stations. Rubber tubing connected all stations to inclined draft gages mounted on the instrument panel.

In addition to the test arrangement shown in Figs. 1 and 2, which has been designated *remote* to signify that the fan is not discharging directly into the plenum, several other test arrangements were utilized. A *close coupled* fan position was produced by removing the five and one-half feet of trunk duct, including the egg-crate straightener and the resistance plate; then reconnecting the equipment. Several other test arrangements were provided by changing the locations of outlets Nos. 1 and 5 along the sides of the plenum. In addition, further variation of conditions was possible by completely closing off any combination of outlets; therefore tests could be run with from one to six outlets open.

When any of the foregoing test arrangements was used, one of the five entrance sections shown in Fig. 3 was located at the plenum entrance, that is, just downstream of Station 2 (see Fig. 1). The principal details of these five entrance sections are:

Type 1—Conventional method of introducing air into a plenum, that is, an abrupt expansion condition. The performance of this section was used as the basis for evaluating the performance of the other types.

Type 2—A diverging section 18 in. long. This length was selected as a maximum distance external to the plenum that would be available for corrective devices in many practical applications.

Type 3—The *Type 2* section with seven full-depth splitter vanes installed.

Type 4—An internal expansion ring having five vanes, the entire device being located inside the plenum.

Type 5—An internal expansion section obtained by inscribing a semicircle about the upstream end of the *Type 3* section and placing it inside the plenum.

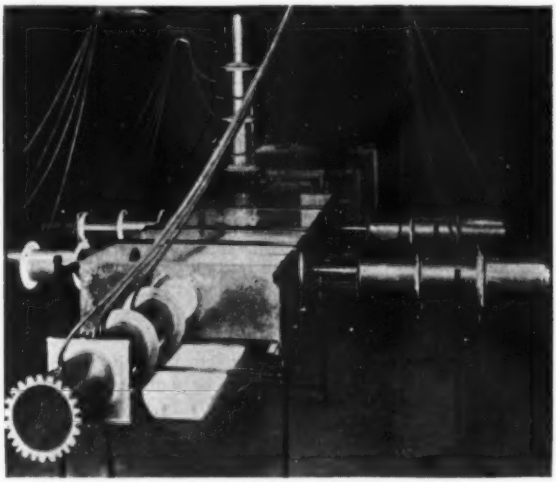

FIG. 2. PHOTOGRAPH OF LABORATORY APPARATUS FOR BOX PLENUM TESTS

CALIBRATION OF EQUIPMENT

The 16 inclined draft gages were calibrated with an Illinois micromanometer[1] to obtain calibration curves for correcting the gage readings to their true values.

The orifices in the orifice box were calibrated with special equipment. When calibrated, any volume between 80 and 2000 cfm entering the system was determined from the static pressure reading at Station 0. For this volume range, the velocities in the trunk duct leading to the plenum were between 64 and 1600 fpm.

The orifices in the branch outlets were calibrated individually with the branch outlets in their usual positions around the box plenum. During calibration of any one outlet, the other outlets were closed off and the air volume flowing was determined by means of the orifice box. Any outlet volume between 80 and

[1] Exponent numerals refer to References.

350 cfm (300-1310 fpm) could then be determined from the pressure differential across the orifice. In addition, the static pressure at the upstream tap (*Example:* Station 51 of Outlet No. 5 in Fig. 1) was recorded during calibration to determine an outlet resistance calibration curve for each outlet with the damper *removed.*

GENERAL TEST PROCEDURE

For any given test arrangement, the total volume was varied over the range of 100 to 350 cfm per outlet in about six steps. Before each test, calculations were made of the correct gage readings required to furnish the respective total volume desired for each step. Each of these volumes was then obtained by

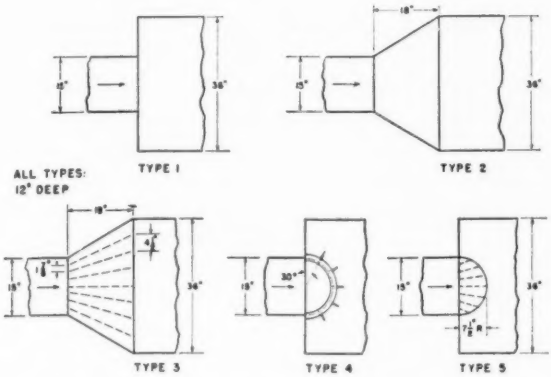

FIG. 3. PLAN VIEW OF ENTRANCE SECTIONS STUDIED

adjusting the fan speed until the desired value of the static pressure at Station 0 was established. The static pressures of Stations 0, 2, 11, 21, 31, 41, 51, and 61, and the pressure differentials across the outlet orifices were recorded.

Since one or more of four standard test conditions were produced in a majority of the various test series, these four conditions have been designated as *basic tests* and are as follows:

Basic Test 1—Unbalanced; all dampers removed. The equivalent length of each outlet was then 54 ft with 175 cfm flowing.

Basic Test 2—Balanced to obtain equal outlet volumes. Modification of *Basic Test 1* with minimum dampering, that is, at least one outlet left as in the *Basic Test 1.*

Basic Test 3—Unbalanced; additional equivalent length of 46 ft added to each outlet by means of the damper. The total equivalent length of each outlet was then 100 ft with 175 cfm flowing.

Basic Test 4—Balanced to obtain equal outlet volumes. Modification of *Basic Test 3* with minimum dampering, that is, at least one outlet left as in the *Basic Test 3.*

TABLE 1. BOX PLENUM TESTS

TEST SERIES	FAN POSITION	ENTRANCE SECTION	TYPES OF TESTS	NUMBER OF OUTLETS OPEN	REMARKS[a]
A	Special	Special	Calibration	One	Calibration of orifice box and outlet orifices.
B	Remote	Type 1	Basic 1, 2, 3, 4	Six	Rotation occurred. Duplicate tests for C and CC flow required.
C	Close Coupled	Type 1	Basic 1, 2	Six	Rotation occurred. Duplicate tests for C and CC flow required.
D	Remote	Type 2	Basic 1, 2, 3, 4	Six	Rotation occurred. Duplicate tests for C and CC flow required.
E	Remote	Type 3	Basic 1, 2, 3, 4	Six	No rotation.
F	Remote	Type 4	Basic 1, 2, 3, 4	Six	Rotation occurred. Duplicate tests for C and CC flow required.
G	Remote	Type 5	Basic 1, 2, 3, 4	Six	No rotation.
H	Remote	Type 5	Basic 1, 2	Six	No rotation. Outlet No. 1 located 7 in. from upstream corner of plenum.
I	Remote	Type 5	Basic 1, 2	Six	No rotation. Outlet No. 1 located 15½ in. from upstream corner of plenum.
J	Remote	Type 5	Basic 1, 2	Six	No rotation. Outlet No. 1 located 11 in. from upstream corner of plenum.
K	Remote	Type 5	Basic 1, 2	Six	No rotation. Outlet No. 1 located 21 in. from upstream corner oi plenum.
L	Remote	Type 5	Basic 1, 2	Six	No rotation. Outlet Nos. 1, and 5 located 7 in. from upstream corners of plenum.
M	Remote	Type 5	Basic 1, 2	Varied	No rotation.
N	Remote	Type 5	Special	Six	Three separate piezometer rings formed by Station 3, 4, 11, and 12; Station 5 and 10; and Station 6, 7, 8, and 9.
O	Remote	Type 5	Basic 1	Varied	No rotation.
P	Remote	Type 5	Basic 1	Varied	No rotation.
Q	Remote	Type 5	Basic 1 and Special	Varied	No rotation. Piezometer ring formed by Station 6, 7, 8, and 9.

[a] Rotation—For several tests, the entering air flowed down one side of the plenum instead of expanding and completely filling the plenum section near the entrance. This condition was designated as Clockwise (C) or Counterclockwise (CC) Rotation, depending upon the direction of the air flow around the plenum as viewed from above. For example, when CC rotation occurred, the air stream from the entering duct fed Outlets Nos. 1, 2, 3, 4, and 5 in order.

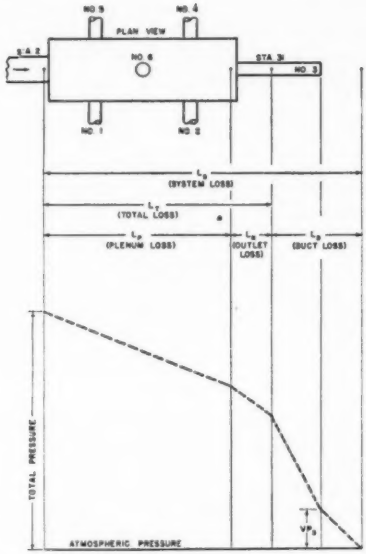

FIG. 4. DESCRIPTION OF LOSSES OCCURRING IN BOX PLENUM

The test series completed, together with supplementary information, are shown in Table 1.

ANALYSIS OF LOSSES

The system pressure loss in any arrangement consisting of a box plenum and ducts is composed of the

1. Entrance loss as the air enters the plenum.
2. Turbulence loss within the plenum.
3. Outlet loss as the air flows from the plenum into the branch outlet.
4. Duct loss, including registers.

For purposes of this investigation, Station 2 (see Fig. 4) represented the beginning of the system under consideration; therefore, when the total pressure at Station 2 above atmospheric pressure was evaluated for any test it represented the system loss for that test arrangement, namely, the sum of the losses of items 1 through 4.

The sum of the pressure losses in items 1, 2, and 3 can be considered as representing the *total loss* of the plenum, and can be evaluated by subtracting the duct loss (item 4) from the total pressure measured at Station 2. This duct

loss (item 4) was determined from the total pressure at Station n1.* Since duct losses in general can be determined by methods in common use, the duct losses of the branch outlets were incidental to the main objectives of this investigation. The total loss was then expressed as

$$L_t = (TP)_o - (TP)_{n1} \quad \ldots \ldots \ldots \ldots \quad (1)$$

in which

L_t = total loss of the plenum, sum of items 1, 2, 3.
$(TP)_o$ = total pressure at Station 2, sum of items 1, 2, 3, 4.
$(TP)_{n1}$ = total pressure at Station n1, item 4.

By methods discussed in detail in Appendix A, it was possible to evaluate the outlet loss (item 3), one of the three components in the total loss as determined from Equation 1. The outlet loss could be expressed as

$$L_n = 0.45(VP)_n \quad \ldots \ldots \ldots \ldots \quad (2)$$

in which

L_n = outlet loss at the entrance into any branch duct, item 3.
$(VP)_n$ = velocity pressure at Station n1.

When this outlet loss, Equation 2, was subtracted from the total loss of the plenum, Equation 1, the result was defined as the *plenum loss.* The plenum loss, which was the sum of the entrance and turbulence losses, could therefore be expressed as

$$L_p = L_t - L_n \quad \ldots \ldots \ldots \ldots \quad (3)$$

in which

L_p = plenum loss, sum of items 1 and 2.

An investigation of the factors affecting the entrance and turbulence losses resulted in the conclusion that they depend upon each other and are not logically separable. The investigation consisted, therefore, in evaluating the plenum loss L_p and the outlet loss L_n for a large number of arrangements of the plenum and branch outlets.

Since the six branch outlets were similar in construction, they were expected to deliver equal volumes. However, when a balanced distribution was not obtained, the dampers in the branch outlets were used to equalize the flow from each outlet. Thus the box plenum system was balanced in exactly the same manner as any air distribution system would be balanced in the field. The additional pressure required to balance the system was designated as the *balancing loss.* This was represented by the *difference* in pressure requirements for balanced and unbalanced tests, designated as *Basic Tests 2* and *1,* respectively. Although the balancing loss would ordinarily be considered as part of the duct loss (item 4), it can be appropriately considered as part of the plenum loss, since the amount of additional pressure required for balance depends upon the pressure loss and air flow characteristics of the plenum.

As will be discussed in a later section of this paper, the pressure loss in the

* n is a general symbol for the number assigned to any outlet for reference purposes only. *Example:* for Outlet No. 2, *Sta. n1* would be written *Sta. 21.*

plenum, including the balancing loss, was materially affected by the design of the expansion section at the entry of the box plenum. Since the losses to be evaluated for *balanced* test conditions included the balancing loss, the results of *Basic Test 2* (the balanced tests) were expected to provide a better index of the performance of the box plenum than the results of *Basic Test 1* (unbalanced tests).

The losses shown in the foregoing equations were measured in inches of water; consequently, any given loss was necessarily associated with a particular volume or velocity. Therefore, the losses were more conveniently expressed as the *number of velocity heads lost* by dividing the loss in inches of water by the velocity head corresponding to the velocity flowing, also in inches of water. This number, being dimensionless, was then independent of any volume or velocity. For example, if Equation 2 is divided by $(VP)_n$, the number of velocity heads lost is 0.45; and this value is applicable for any volume flowing in the outlet.

PERFORMANCE WITH ABRUPT EXPANSION ENTRANCE SECTION (TYPE 1)

a. Plenum Losses. The results of the *Basic Tests 1* and *2* of Series B indicated that the plenum loss was 1.17 velocity heads in the inlet duct for unbalanced conditions and was increased to 1.77 velocity heads in the inlet duct when the system was balanced for equal flow from each outlet.

The plenum loss under balanced conditions is more than *five times* the abrupt expansion entrance loss obtained from current literature.[2, 3] Since no published data are available for evaluating the turbulence loss within a plenum, the entire plenum loss is generally assumed to be represented by the entrance loss alone. The results of this investigation show clearly that this is a dangerous assumption, because the entrance loss obtained from the literature represented less than 20 percent of the actual loss which occurred within the plenum.

b. Rotational Flow. During the test series conducted with the *Type 1* entrance section and with the fan in both the remote and close coupled positions, the silk directional indicators showed a definite rotation of the air within the plenum in either the clockwise or counterclockwise direction. The particular direction of rotation which would occur could not be predicted in advance of a test.

The effect of this rotation on the distribution of air to the outlets was of considerable importance, because the amount of air going to each outlet depended upon the location of the outlet with respect to the rotating stream. Moreover, the flow could be readily changed from clockwise to counterclockwise rotation, or vice versa, thus upsetting the former distribution. For example, Outlet No. 1 delivered more air than Outlet No. 5 when counterclockwise rotation occurred in the *Basic Test 1* (unbalanced) tests. However, when Outlet No. 1 was closed off for 5 to 20 sec, the direction of rotation shifted to clockwise and was stable in that direction, even when Outlet No. 1 was reopened. Outlet No. 1 then delivered only 80 percent of its former volume.

Because of the unequal distribution of the air to the outlets during the unbalanced tests, high resistances had to be imposed upon the favored outlets with the dampers to bring the system into balance at equal outlet volumes; consequently the system had a high balancing loss. Moreover, when the system was balanced for a given direction of rotation, it would become unbalanced if the direction of rotation changed.

With respect to the pressure losses, tests indicated that they were the same irrespective of the direction of rotation. Furthermore, the distribution of the air to the outlets was *symmetrical;* for example, if Outlets Nos. 1 and 5 (see Fig. 1)

delivered 180 and 150 cfm, respectively, during counterclockwise rotation, they would deliver 150 and 180 cfm, respectively, during clockwise rotation.

The impossibility of obtaining reliable design data for such an unstable system is apparent. Furthermore, the use of the *Type 1* entrance section in field applications should be avoided, since the balance could conceivably be upset by stopping and restarting the fan or temporarily closing off a register.

c. Effect of Adding Equal Resistance to Each Outlet. The purpose of the *Basic Tests 3* and *4* was to determine the effect of adding equal resistance to each outlet

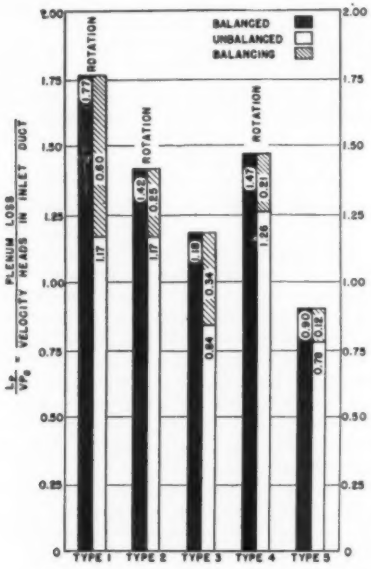

FIG. 5. PLENUM LOSSES WITH FIVE TYPES OF ENTRANCE SECTIONS

upon the pressure loss and air flow characteristics of the box plenum. The results of these tests were compared with those of *Basic Tests 1* and *2*. This comparison showed that the total pressure at Station *2* was higher by an amount equivalent to the resistance added. However, the calculated total losses were the same for the *Basic Tests 1* and *3*, and also for the *Basic Tests 2* and *4*. It was therefore concluded that the total loss of the plenum was unaffected by the addition of equal resistances to each outlet. Also, the added resistance had no perceptible effect upon the air flow characteristics: the individual outlet volumes were comparable and rotational flow occurred.

d. Effect of Fan Discharging Directly Into Plenum. The Series C tests were conducted with the close coupled test arrangement in order to investigate the effect of the fan discharging directly into the plenum. The previous tests with the remote fan position (Series B) were duplicated to provide direct comparison of results. The system loss with the fan in the close coupled position was slightly greater than

the corresponding loss in the remote position. However, for all practical purposes, the losses can be considered the same for either test arrangement. Furthermore, the distribution of the air to the outlets was the same, and rotational flow occurred during both test series.

COMPARATIVE PERFORMANCE OF FIVE ENTRANCE SECTIONS

Since the use of the abrupt expansion section *(Type 1)* resulted in high losses and unstable flow conditions, four additional entrance sections (Fig. 3) were developed and tested. The plenum losses are shown in Fig. 5. These results were obtained from the *Basic Tests 1* and *2* with all six outlets open.

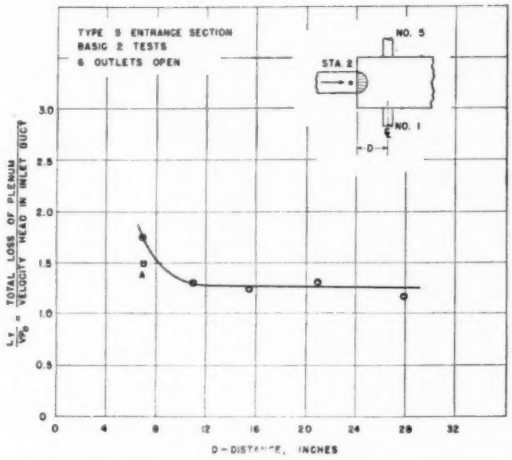

FIG. 6. EFFECT OF OUTLET LOCATION ON PRESSURE LOSS OF BOX PLENUM

It may be noted that rotation occurred with *Types 1, 2,* and *4.* As previously discussed, the use of any entrance section which results in rotational flow should be avoided.

The effect of adding equal resistances to each outlet was also investigated with *Types 2, 3, 4,* and *5.* The results indicated that the plenum loss alone was not affected. This result agreed with that obtained with the *Type 1* section.

The following factors were selected for evaluating the performance of any entrance section:

1. The plenum loss under balanced conditions.
2. The additional pressure required to balance the system.
3. The stability of the system, or the ability to maintain fixed flow rates.
4. The space requirements external to the plenum.

The *Type 5* entrance section was superior to the others in all of these respects, and was therefore studied in greater detail.

PERFORMANCE WITH TYPE 5 ENTRANCE SECTION

A number of test series were conducted with the *Type 5* section to investigate the effect of moving one or more outlets, and the results are presented in Fig. 6. The total loss of the plenum increased considerably when the outlet was located closer than approximately 12 in. from the corner of the plenum. Therefore, the centerline of any outlet should be more than 12 in. from the corner if excessive pressure losses are to be avoided.

An additional test series was conducted with both Outlets Nos. 1 and 5 located seven inches from the corner of the plenum. As indicated by point A in

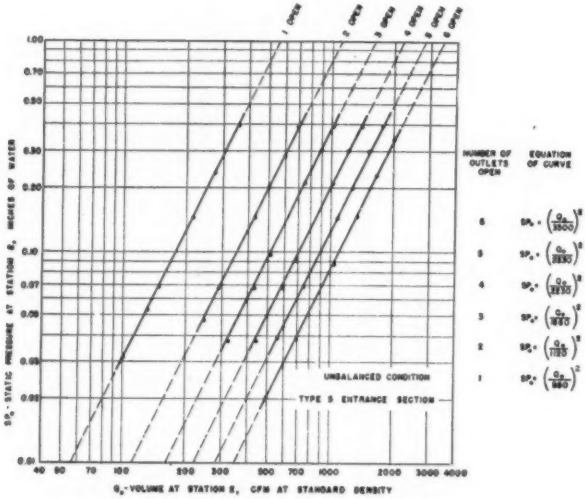
NUMBER OF OUTLETS OPEN
EQUATION OF CURVE
UNBALANCED CONDITION
TYPE 5 ENTRANCE SECTION
Q_s - VOLUME AT STATION 2, CFM AT STANDARD DENSITY

FIG. 7. VARIATION OF STATIC PRESSURE AND VOLUME AT STATION 2 WITH NUMBER OF OUTLETS OPEN

Fig. 6, the total loss is less under these conditions. This indicates that a symmetrical location of outlets is desirable when a distance D of less than 12 in. must be used. For a distance D greater than 12 in., the performance of the system is not affected by the relative locations of outlets.

The effect of varying the number of outlets open was also investigated. The relationships of the static pressures and air volumes are shown in Fig. 7. Since for any given number of outlets open numerous combinations were possible, several representative combinations were tested for each condition. The static pressure requirements were found to be independent of the particular outlets open in any one combination.

From the data of Fig. 7 and the equations developed in Appendix B, the curve of Fig. 8 was derived. The plotted points, as compared with the curve, represent deviations of less than three percent in the test data.

The equation of this curve, which represents the *unbalanced flow condition,* can also be expressed

$$L_t = 0.80(VP)_o + 0.45(VP)_n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which

L_t = total loss of the plenum, inches of water.
$(VP)_o$ = velocity head at Station 2, inches of water.
$(VP)_n$ = velocity head at Station nl, inches of water.

The total loss, therefore, can be expressed as the sum of a function of the velocity head in the entrance duct and a function of the velocity head in the

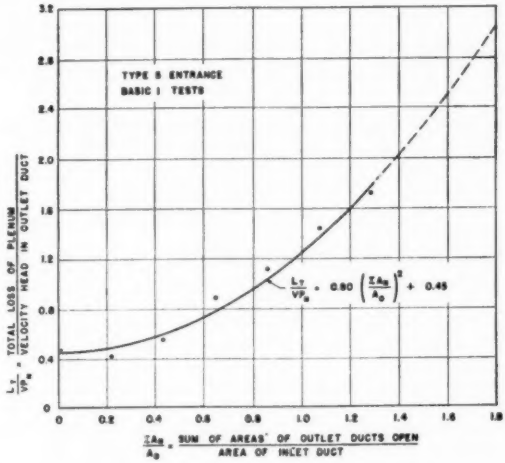

FIG. 8. TOTAL LOSS OF PLENUM WITH VARIABLE RATIOS OF OUTLET TO INLET AREA

outlet duct. The term $0.80(VP)_o$ represents the plenum loss L_p, and the term $0.45(VP)_n$ represents the outlet loss L_n. The plenum loss includes the entrance and turbulence losses; however, as previously stated, these two losses depend on each other and cannot be logically separated.

For the *balanced flow condition,* an equation similar in form to Equation 4 can be derived. In this connection, the balancing loss with the *Type 5* section (see Fig. 5) was extremely small with all six outlets open. Moreover, as the number of outlets open was reduced, the balancing loss became progressively less—in fact, it was necessarily zero when only one outlet was open. Therefore, the data obtained with all outlets open were used to derive an expression for the total loss for the balanced flow condition, as follows:

$$L_t = 0.90(VP)_o + 0.45(VP)_n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

This equation is recommended for use with the *Type 5* entrance section, *provided* no outlet is located within 12 in. of the upstream corners of the plenum.

Equations 4 and 5, which were derived from curves having dimensionless ordinates, have a definite possibility of being extrapolated to other sizes of inlet and outlet ducts and to larger numbers of outlets open. Additional tests are in progress to investigate this possibility.

The favorable performance characteristics of the *Type 5* entrance section can be summarized as follows:

1. The plenum loss under balanced conditions was only 51 percent of that for the *Type 1* section.
2. The additional pressure required to balance the system was very small.
3. The stability of the system was excellent, since no rotational flow occurred or could be made to occur.
4. The space requirement was a minimum, since the entrance section occupied no space external to the plenum.
5. The section was easily fabricated and installed.

Because of these favorable characteristics, this section is recommended for use, and the necessary construction details are shown in Fig. 9.

General Equation for Total Loss

The relations expressed in Equations 4 and 5 can be used to develop a general equation for the total loss of a box plenum, which is

$$L_t = K_o(VP)_o + K_n(VP)_n \quad \ldots\ldots\ldots\ldots (6)$$

in which

L_t = total loss of any box plenum, inches of water.
K_o = a constant which depends on factors such as the entrance section, amount of turbulence within the plenum, and the physical dimensions of the plenum.
K_n = a constant which depends upon the performance of the plenum take-off fitting.
$(VP)_o$ = velocity head in the entrance duct, inches of water.
$(VP)_n$ = velocity head in any outlet duct, inches of water.

For the branch outlets used in this investigation, K_n has been evaluated as equal to 0.45. For the five entrance sections investigated, the values of the balanced plenum losses shown in Fig. 5 are also values of K_o. Therefore, the total loss equations for the five types of entrance sections, for balanced flow conditions, are as follows:

$$\textit{Type 1}—L_t = 1.77(VP)_o + 0.45(VP)_n \quad \ldots\ldots\ldots (7)$$

$$\textit{Type 2}—L_t = 1.42(VP)_o + 0.45(VP)_n \quad \ldots\ldots\ldots (8)$$

$$\textit{Type 3}—L_t = 1.18(VP)_o + 0.45(VP)_n \quad \ldots\ldots\ldots (9)$$

$$\textit{Type 4}—L_t = 1.47(VP)_o + 0.45(VP)_n \quad \ldots\ldots\ldots (10)$$

$$\textit{Type 5}—L_t \doteq 0.90(VP)_o + 0.45(VP)_n \quad \ldots\ldots\ldots (5)$$

In order to investigate the applicability of Equation 6 to other sizes and

types of plenums, additional tests are in progress with a three-foot long box plenum.

CONCLUSIONS

1. The total pressure loss associated with a box plenum is composed of an entrance, turbulence, and outlet loss. The entrance and turbulent losses cannot be logically separated.
2. The pressure loss at the entrance to the branch outlets is in good agreement with the generally accepted value.
3. The pressure losses and air flow conditions within a box plenum are not affected by the addition of equal resistance to each outlet.
4. For all practical purposes, the losses with the remote and close coupled fan positions are the same. Therefore, the results will apply to field conditions when the fan is located very close to the plenum.
5. Take-off fittings for branch outlets should not be located within 12 in. of the upstream corners of the plenum if excessive balancing losses are to be avoided.

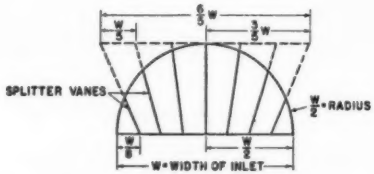

FIG. 9. CONSTRUCTION DETAILS OF TYPE 5 ENTRANCE SECTION (PLAN VIEW)

6. The plenum loss with the abrupt expansion entrance section *(Type 1)* installed is much higher than the calculated entrance loss alone, indicating the loss due to turbulent mixing in the plenum is severe.
7. Unless the entrance section of a box plenum is carefully designed, large pressure losses and unstable distribution of air to the branch outlets will result.
8. When the *Types 1, 2,* and *4* entrance sections were used, rotational flow occurred and the system was unstable. The use of these three types should be avoided.
9. The *Type 5* entrance section is superior to the other types tested from the standpoint of total loss, pressure required to balance the system, stability of the system, and space requirements. This section is recommended for use in the field. The plenum loss with this section can be taken as 0.90 velocity heads in the inlet duct, provided no branch take-off is located within 12 in. of the upstream corners of the plenum.

ACKNOWLEDGMENT

On October 1, 1947, the *American Gas Association,* Committee on Domestic Gas Research, as part of the Association's comprehensive research program (Par Plan, *AGA Project DGR-2-AC*) entered into an agreement for cooperative research with the Engineering Experiment Station at the University of Illinois,

the objective being the improvement of air distribution systems for gas year 'round air conditioning. This paper is a progress report of the initial laboratory investigations of box-type plenums. The laboratory work represents one phase of the overall research program.

Acknowledgment is made to the *AGA* Technical Advisory Group for Gas Summer Air Conditioning Research, for the advice and counsel received, and to W. K. Ender and G. W. Wilson, former graduate students at the University, for services rendered in collecting and reducing test data. These results will ultimately comprise part of a bulletin of the University of Illinois Engineering Experiment Station.

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APPENDIX A

EVALUATION OF THE OUTLET LOSS

Since all outlets were butted to the sides of the plenum, an abrupt contraction loss occurred as the air flowed from the plenum into any outlet. The results of the Series A and Series Q tests for the condition of one outlet open provided two separate methods for evaluating this loss.

The outlet loss based on the Series A tests was obtained by assuming that the outlet loss was equal to the total loss L_t. Since, with only one outlet open, the velocity head in the branch outlet was 21.8 and 126.0 times the velocity heads in the entrance duct (Station 2) and plenum, respectively, this assumption represented a good approximation of the actual outlet loss.

With regard to the second method, one of the Series Q tests was conducted with only Outlet No. 3 open and Stations 6, 7, 8, and 9 connected in parallel to form a piezometer. With this piezometer, the total pressure existing just ahead of the entrance to Outlet No. 3 was obtained. The difference between this total pressure and the total pressure at Station 31 (see Fig. 4) represented the contraction loss occurring at the entrance to the outlet.

Since the pressure differences were small, and thus the potential error large, the average of the values determined by the two methods was selected. The final result obtained was an outlet loss equal to 0.45 of the velocity head existing in the branch outlet.

APPENDIX B

COMPUTATION OF RESULTS

Energy Leaving the Plenum

Since all outlets were made alike, they had nearly identical resistance calibration curves. Within the limits of accuracy possible, the static pressure at the upstream tap of any outlet (Station nl) could be expressed by a single equation of the form

$$(SP)_{n1} = \left(\frac{Q_n}{K}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (11)$$

in which

$(SP)_{n1}$ = static pressure at the up-stream tap of any outlet, inches of water.
Q_n = volume flowing in any outlet as obtained from the orifice differential calibration curves, cubic feet per minute of air at standard density.
K = a constant of proportionality.
n = subscript denoting the number of any outlet concerned. *Example:* for Outlet No. 2, Equation 11 becomes

$$(SP)_{21} = \left(\frac{Q_2}{K}\right)^2$$

Since logarithmic coordinate paper was used, the constant K was easily evaluated and Equation 11 became

$$(SP)_{n1} = \left(\frac{Q_n}{710}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (12)$$

In order to evaluate the total energy leaving the plenum, it was necessary to consider the total pressure (TP), which is the sum of the static (SP) and velocity (VP) pressures. Thus, for any outlet

$$(TP)_{n1} = (SP)_{n1} + (VP)_{n1} \quad \ldots\ldots\ldots\ldots \quad (13)$$

Also, for any outlet

$$Q_n = A_n V_n \quad \ldots\ldots\ldots\ldots \quad (14)$$

or, for air at standard density and 7 in. diameter outlets,

$$Q_n = (0.267)\ 4005\sqrt{(VP)_n} \quad \ldots\ldots\ldots\ldots \quad (15)$$

in which

$(VP)_n$ = velocity head in any outlet corresponding to the mean velocity of flow, inches of water.

When solved for $(VP)_n$, Equation 15 became

$$(VP)_n = \left(\frac{Q_n}{1070}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (16)$$

Equations 12 and 16 were substituted into Equation 13 to obtain

$$(TP)_{n1} = \left(\frac{Q_n}{710}\right)^2 + \left(\frac{Q_n}{1070}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (17)$$

which was combined into one term. The final result was

$$(TP)_{n1} = \left(\frac{Q_n}{592}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (18)$$

which expressed the total pressure, in inches of water, at Station nl as a function of the volume flowing in the outlet, provided only that the outlet was *not* dampered.

Energy Entering the Plenum

An expression for the total pressure at Station 2 was obtained by an analysis similar in every respect to the foregoing. At Station 2, the 12 in. $\times$ 15 in. trunk duct had an area of 1.25 sq ft. Thus, Equation 14 became

$$Q_o = A_o V_o \quad \ldots \ldots \ldots \ldots \ldots \quad (19)$$

where the subscript refers to Station 2. Equation 19 led directly to

$$(VP)_o = \left(\frac{Q_o}{5000}\right)^2 \quad \ldots \ldots \ldots \ldots \ldots \quad (20)$$

in which

$(VP)_o$ = velocity head in trunk duct corresponding to the mean velocity of flow, inches of water.

An equation similar in form to Equation 17 was then

$$(TP)_o = \left(\frac{Q_o}{K_{SP}}\right)^2 + \left(\frac{Q_o}{5000}\right)^2 \quad \ldots \ldots \ldots \quad (21)$$

in which

K_{SP} = a constant of proportionality to be determined experimentally. (*Examples:* equations of curves shown in Fig. 7.)

After K_{sp} was determined experimentally, Equation 21 was reduced to

$$(TP)_o = \left(\frac{Q_o}{K_{TP}}\right)^2 \quad \ldots \ldots \ldots \ldots \ldots \quad (22)$$

in which

K_{TP} = a *total pressure* constant of proportionality which included the two constants in Equation 21.

Equation 22 expressed the total pressure, in inches of water, at Station 2 as a function of the total volume entering the plenum. Clearly, K_{TP} varied with the conditions of the test, such as the type of expansion section used and the number of outlets open.

Evaluation of Losses

Since the difference in energy entering and leaving the plenum represented the loss, Equations 18 and 22 provided the necessary relationships for determining the total pressure loss from Station 2 to Station nl for any *undampered* outlet. Thus, Equation 1 became

$$L_t = \left(\frac{Q_o}{K_{TP}}\right)^2 - \left(\frac{Q_n}{592}\right)^2 \quad \ldots \ldots \ldots \ldots \quad (23)$$

This equation applied to each outlet for the tests conducted with all dampers removed, that is, *unbalanced flow conditions (Basic Test 1).* However, since the outlet volumes were not the same, the total loss depended upon the particular outlet volume selected. Therefore, an average total loss was required. In this connection, the assumption that

$$Q_n = \overline{Q_n} \quad \ldots \ldots \ldots \ldots \ldots \quad (24)$$

in which

$\overline{Q_n}$ = average outlet volume, cubic feet per minute of air at standard density

introduced a maximum error of 0.7 percent. Since this error was well within the limits of accuracy of the investigation, Equation 23 was written

$$L_t = \left(\frac{Q_o}{K_{TP}}\right)^2 - \left(\frac{\overline{Q_n}}{592}\right)^2 \quad \ldots \ldots \ldots \ldots \quad (25)$$

Since

$$Q_o = N\,\overline{Q}_n \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (26)$$

in which

N = number of outlets open during a given test.* *Example:* for any combination of three outlets open, $N = 3$.

Substitution of Equation 26 in Equation 25 resulted in a general equation for the total loss with any number of outlets open.

$$L_t = \left(\frac{N\,\overline{Q}_n}{K_{TP}}\right)^2 - \left(\frac{\overline{Q}_n}{592}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (27)$$

In order to express Equation 27 in a dimensionless form, it was divided by Equation 16 to obtain

$$\frac{L_t}{(VP)_n} = \frac{\left(\frac{N\,\overline{Q}_n}{K_{TP}}\right)^2 - \left(\frac{\overline{Q}_n}{592}\right)^2}{\left(\frac{Q_n}{1070}\right)^2} \quad \ldots\ldots\ldots\ldots \quad (28)$$

which, when reduced to its final form,

$$\frac{L_t}{(VP)_n} = \left(\frac{1070\,N}{K_{TP}}\right)^2 - 3.27 \quad \ldots\ldots\ldots\ldots \quad (29)$$

was used to calculate the total loss of the plenum for all *Basic Test 1* tests.

During the *Basic Test 2* tests, *balanced flow conditions* were obtained by adjusting the dampers until the outlet volumes were equal. Therefore, any outlet volume Q_n was also the average outlet volume $\overline{Q}_n$; moreover, at least one outlet was *not dampered* during the *Basic Test 2* tests. Since these were precisely the conditions used in the derivation of the given equations for the *Basic Test 1* tests, the foregoing equations were also applicable to the one or more outlets not dampered during the *Basic Test 2* tests. Furthermore, since whatever resistance was added to balance the system was charged against the system performance, the total loss computed from Equation 29 for any one undampered outlet represented the total loss of the system under *balanced conditions* of flow. Therefore, Equation 29 was used to calculate the total loss of the plenum for all *Basic Test 2* tests.

When Equations 2 and 3 were divided by $(VP)_n$ to obtain dimensionless expressions, the plenum loss in velocity heads in the outlet duct was easily evaluated with the aid of Equation 29.

A relation between the velocity heads existing in the inlet and outlet ducts was derived as follows: Equations 19 and 26 were equated to obtain

$$A_o V_o = N\,\overline{Q}_n \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (30)$$

or, with the aid of Equations 14 and 24,

$$A_o V_o = N\,A_n V_n \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (31)$$

For air at standard density and the respective areas, Equation 31 became

$$(1.25)\,(4005)\,\sqrt{(VP)_o} = N\,(0.267)\,(4005)\,\sqrt{(VP)_n} \quad \ldots\ldots \quad (32)$$

* This symbol should not be confused with the symbol "n" which refers to the number assigned to any outlet for reference purposes only.

which, when rationalized and rearranged, provided the desired result

$$(VP)_n = \left(\frac{21.8}{N^2}\right)(VP)_o \quad \ldots\ldots\ldots\ldots \quad (33)$$

Thus, any loss could be expressed in terms of either the velocity head in the outlet duct or the velocity head in the inlet duct.

Derivation of Equation in Fig. 8

The curve of Fig. 8 and its equation were established as follows: The total losses were plotted against the number of outlets open. However, any given number of outlets open could be expressed as a ratio of the inlet and outlet duct areas. When this was done, the completely dimensionless plot of Fig. 8 was obtained.

The problem was then the relating of the plotted points to a single curve. In this connection, an attempt was made to fit the plotted points with the simplest relation which might hold. This was

$$L_t = K_o(VP)_o + K_n(VP)_n \quad \ldots\ldots\ldots\ldots \quad (34)$$

where K_o and K_n are constants. Using the value of the outlet loss determined by the methods discussed in Appendix A, Equation 34 became

$$L_t = K_o(VP)_o + 0.45(VP)_n \quad \ldots\ldots\ldots\ldots \quad (35)$$

Next, Equation 33 was used to eliminate $(VP)_o$ and obtain

$$L_t = K_o\left(\frac{N^2}{21.8}\right)(VP)_n + 0.45\,(VP)_n \quad \ldots\ldots\ldots \quad (36)$$

which, when divided by $(VP)_n$, became

$$\frac{L_t}{(VP)_n} = K_o\left(\frac{N^2}{21.8}\right) + 0.45 \quad \ldots\ldots\ldots\ldots \quad (37)$$

However, since the identity

$$\frac{N^2}{21.8} = \left(\frac{\Sigma A_n}{A_o}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots \quad (38)$$

could be written, Equation 37 was expressed

$$\frac{L_t}{(VP)_n} = K_o\left(\frac{\Sigma A_n}{A_o}\right)^2 + 0.45 \quad \ldots\ldots\ldots\ldots \quad (39)$$

Using trial values of K_o, Equation 39 was calculated and plotted on Fig. 8 until a value of K_o providing the best fit was obtained. The final result was

$$\frac{L_t}{(VP)_n} = 0.80\left(\frac{\Sigma A_n}{A_o}\right)^2 + 0.45 \quad \ldots\ldots\ldots\ldots \quad (40)$$

which is the equation of the curve shown in Fig. 8. The plotted points, as compared with the curve, were found to represent deviations of less than three percent in the test data. Since three percent was within the limits of accuracy expected, it was concluded that the relations expressed by Equations 34 and 40 provided a satisfactory fit of the plotted points.

DISCUSSION

F. L. MEYER and J. J. BOLAND*, Peoria, Ill. (WRITTEN): The authors are to be commended for the detailed study of the air flow characteristics within a box plenum. Obviously, a great deal of careful thought and effort have gone into the making of this report and the practicability of the proposed method is gratifying. The use of a box plenum, in which the results were assured to be stable, would be of great assistance in design of overhead supply ducts or central air distributing compartments in the applications mentioned.

According to Table 1, assured results are dependent on the use of Type 3, or its modified version Type 5, to obtain non-rotational flow within the box plenum. It would appear that the tests with these entrance sections were not carried far enough to assure stability in all cases. Tests were conducted on Type 1, both remote and close-coupled, and the results indicated that the characteristics were very similar. To make the close-coupled test, it was necessary to remove the egg-crate straighteners and the resistance plate. Both of these items help to produce a uniform velocity contour as the air approaches the entrance section. It would be interesting to know if an investigation was made of the velocity contour at Station 2 to show whether the velocity contour was uniform and the rotation in the box plenum was produced by the rapid expansion of air into the box plenum. The successful use of the entrance section, Type 3 or Type 5, would appear to be dependent upon the uniform velocity contour approaching the entrance section produced by the egg-crate straighteners or the resistance plate. If the velocity contour approaching the entrance section, Type 3 or Type 5, should not be uniform due to close-coupling of heating or cooling equipment or an elbow upstream of the box plenum, the air flow in the box plenum may also be unstable. If tests prove this to be the case, the use of egg-crate straighteners or a resistance plate may be required to produce a uniform velocity contour. This would impose additional resistance in the system and increase the box plenum loss.

JOHN EVERETTS, JR., Philadelphia, Pa.: I wish to compliment the authors on their excellent presentation of a subject of importance in design of air conditioning and heating systems.

With entrance duct the same depth as plenum, would resistance change with other $\frac{A_1}{A_2}$ ratios and with four sides of entrance duct effective for expansion?

I suggest that work be continued particularly on exhaust plenums and plenum intakes for fans.

C. S. LEOPOLD, Philadelphia, Pa.: Were there any measurements or observations of effect of the deflector on noise?

AUTHORS' CLOSURE: Referring to the comments of Messrs. Meyer and Boland, the velocity contour at Station 2 was determined for several arrangements of the apparatus. During calibration of the orifice box, the apparatus downstream of Station 2 was removed and a transition section and 15 ft of 10-in. diameter duct installed. For this condition the velocity contour at Station 2 was, for all practical purposes, uniform. With the Type 1 section installed as shown in Fig. 1 of the paper, the velocity contour varied with the direction of rotation. During clockwise rotation of the air stream, the velocity contour showed definitely higher velocities in the left-hand side of the trunk duct as we stand at the orifice box and look downstream. During counterclockwise rotation the velocities at points near the right-hand side of the trunk duct at Station 2 were definitely higher than on the left side. The velocity contour was therefore determined to a great extent by the direction of flow in the plenum. Velocity contours at Station 2 were not obtained when the Types 3, 4, and 5 were

* The Meyer Furnace Company.

used because these sections prevented the placement of a pitot tube at Section 2. However, exploratory studies were made with the resistance plate removed and the Type 5 section installed. The pressure losses in the plenum were the same and rotational flow still could not be made to occur. From our experience with the Type 5 section, it is our opinion that stable operation with this section could be expected except possibly for cases of extreme non-uniformity of velocity profile at the entrance to the plenum.

As to Mr. Everetts' questions, the ratio of the plenum area at the entrance to the trunk duct area was 2.4. The use of area ratios between 2 and 3 would change the resistance only slightly. Ratios outside this range would be used only rarely. If the depth of the entrance duct were smaller than that of the plenum, the Type 5 entrance would not normally be used because it provides for expansion only in one direction; that is, for a horizontal expansion of the air stream. An entrance section providing for efficient expansion both horizontally and vertically is difficult to fabricate. We suggest instead that the entrance duct be made the same depth as the plenum wherever possible.

Qualitative observations by ear were made of the noise. Insofar as we could determine, no noise due to the deflector was audible. However, in a residential installation with the air conditioner close-coupled to the box-plenum and with a very short branch duct leading to one of the rooms, equipment noise could be easily transmitted through the duct system. For such a case, acoustical treatment could conceivably be necessary.

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No. 1368

EARTH AS A HEAT SOURCE OR STORAGE MEDIUM FOR THE HEAT PUMP

By E. W. GUERNSEY*, P. L. BETZ** AND N. H. SKAU†, BALTIMORE, MD.

THE FUTURE development of space heating by the heat pump is to a considerable degree dependent on the successful adaptation of available heat-sources. There has been some tendency, when apparent difficulties in the use of air, well water or other heat-sources have been recognized, to look to the use of a heat exchanger buried in the earth as a more promising source. It is the purpose of this paper to examine critically some of the factors which would be expected to determine the practicability of such earth heat-sources. The paper will be primarily concerned with the results of a series of calculations of heat flow under various specified conditions and over extended periods of time when the supply of heat depends solely on heat capacity of the soil and conduction through the soil.

It is believed that heat capacity and thermal conductivity are the most important factors, and that the results may therefore be directly useful as indicating the limitations of ground coils for *general* use. There may be, of course, special coil locations, as for example near ground water streams, where there are important contributions to heat supply not related to heat capacity and heat conduction. In such special cases the calculations of the present paper may be less useful.

ASSUMED CONDITIONS

Solutions are presented in general terms to permit application to various sets of circumstances. In addition, however, detailed calculations for particular assumed conditions are shown in order to give an immediate indication as to the practical outlook. The particular conditions are in general such as to give minimum values for the required heat transfer surface. Thus, the surfaces are assumed to be located in a highly conducting, rather wet soil, (25 to 30 percent moisture) having the properties:[1] k, 0.896; c_p, 0.45; ρ, 103; α, 0.0193.

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† Research Chemist, Consolidated Gas Electric Light and Power Co. of Baltimore.

[1] Exponent numerals refer to References.

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It is assumed further for the sample calculations that 50,000,000 Btu of heat are to be extracted from the earth annually. This corresponds to the case of a small, well-insulated house in an area where the minimum temperature is 0 F, the temperature deficiency is 4500 degree days per season, the total heating requirement is 75,000,000 Btu, and the coefficient of performance of the heat pump is three.

It is generally assumed for the calculations that the temperature of the heat transfer surface may deviate from the normal average earth temperature as much as 40 F.

Several idealized cases may now be considered, these being such as to permit precise solution and at the same time to correspond as nearly as possible to some particular practical situation. This permits the application of the results of the calculations with a minimum of arbitrary adjustment.

CONSTANT HEAT WITHDRAWAL FROM CYLINDRICAL SURFACE

Case 1: Heat is withdrawn at a constant rate at a cylindrical surface in contact with earth of infinite extent and uniform initial temperature. Effects of cylinder ends are ignored; these effects become smaller as cylinder length increases. No account is taken of effects of heat interchange with atmosphere, which are small for the case of a long cylinder extending vertically into the earth.

Newman[2] following the Kelvin concept developed equations for the temperature of continuous sources of either the cylindrical or line type. The line source equation is easier to use and is given below:

$$\theta = \frac{A}{5.4575k} \int_{-\infty}^{\log_{10} \frac{\alpha t}{r^2}} e^{-\frac{r^2}{4\alpha t}} \, d \log_{10} \left(\frac{\alpha t}{r^2} \right) \quad \ldots\ldots\ldots\ldots (1)$$

In the present work calculations were made using both expressions and it was found that they agree within 1 percent for small cylinders after times t greater than $\frac{8r^2}{\alpha}$, *i.e.*, when t>0.5 hr for a ¾-in. pipe and when t>10.5 hr for a 3.82-in. pipe. Errors in Newman's evaluation of the cylindrical source equation were corrected before making the above comparison.

The temperature change, θ_s, at the surface of the cylinder has a special interest, since in the practical use of the cylinder as a heat-source for the heat pump, the design would have to be such that the heat receiving surface would not fall below an acceptable minimum temperature after an extended period. Such temperatures may be found by inserting values for r in Equation 1 corresponding to the radii of the cylinders of the various sizes which may be considered. The equation was solved with the aid of available tables[3] giving the values of a related integral. An empirical expression has been developed giving the value of the integral with negligible error for values $(\alpha t / r^2)$ greater than one. Using this, Equation 1 becomes

$$\theta = \frac{A}{5.4575k} \left[\log_{10} \frac{\alpha t}{r^2} + 0.106 \frac{r^2}{\alpha t} + 0.351 \right] \quad \ldots\ldots\ldots (2)$$

The relations between pipe surface temperature and time, through an initial 20-year period, are shown in the curves of Fig. 1.

It will be seen that at the beginning of the withdrawal of heat from earth, the temperature changes rapidly at the surface of the cylinder, but much less rapidly as time proceeds. It is shown that increasing the cylinder diameter results in a relatively small decrease in the temperature difference between initial ground temperature and cylinder surface temperature.

The application of these curves may now be illustrated. It is assumed that an installation for supplying earth-heat to a heat pump would be considered unsatisfactory unless it is able to maintain the heat supply for at least 20 years. It is indicated in Fig. 1 that the surface of the 3.82 in. diameter tube, (1 sq ft

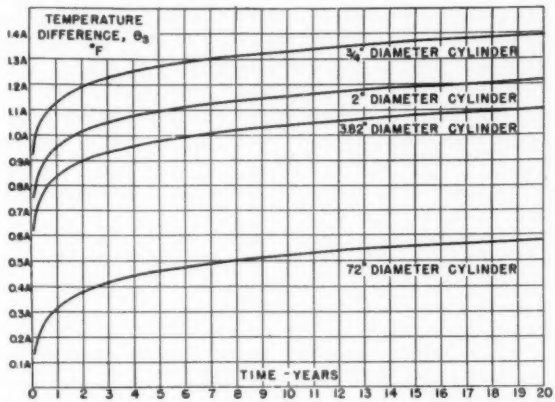

FIG. 1. TEMPERATURE DIFFERENCE BETWEEN SURFACE OF CYLINDERS AND INITIAL EARTH TEMPERATURE AS RELATED TO TIME

Heat absorption rate $= A$ Btu/(hr) (linear ft). Earth constants: $k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

area per linear foot) for example, has dropped at the end of 20 years to 1.1 A deg below the original uniform ground temperature; that is, the drop in degrees is 1.1 times the continuous rate of heat extraction in Btu per hour per linear foot of cylinder.

If it is permissible for the heat absorbing surface to drop at the end of 20 years to a temperature 40 F below the original ground temperature, the corresponding rate of heat withdrawal, A, would be $40/1.1 = 36.3$ Btu per (hour) (foot). If the yearly withdrawal of 50,000,000 Btu were at a uniform rate, that is $50{,}000{,}000/8760 = 5710$ Btu per hour, the indicated length for 20 years of continuous operation would be $5710/36.3 = 157$ ft for the 3.82 in. diameter cylinder. The corresponding lengths indicated for cylinders of the various sizes shown in Fig. 1 are found to be as follows:

Diameter—In.	¾	2	3.82	72
Length for Continuous Withdrawal of 5710 Btu per (hour)—Ft	198	174	157	83

Ingersoll and Plass[4] have shown a similar calculation for continuous heat withdrawal from two soils. The actual lengths required in a practical case would be substantially more than those given here, not only because the soil characteristics, the annual heat requirement and the allowable temperature variation would often be less favorable than assumed, but especially because the yearly heat withdrawal would be largely concentrated in a few winter months and not spread uniformly over the year. The probable magnitude of the correction for the effect of the deviation from constancy in the actual heat withdrawal schedule will be considered at a later point.

In addition to determining the course of the temperature change at the cylinder surface with time, it is possible to determine temperature changes in the soil at various distances from the cylinder surface. For these changes Equations 1 and 2 are applicable, and these relationships are shown in Fig. 2. In Fig. 3 the distribution of temperature through the soil at the end of 20 years is shown.

Such data are pertinent to the determination of space required between two or more cylinders in order that they will not in time interfere with each other. Further analysis of the data in Fig. 3 permits calculation of the portion of the total heat supply for a 20-year period which is drawn from each of the various shells of earth surrounding the cylinder.

The cumulative contributions of the various shells are shown in Fig. 4. The curves of Figs. 2, 3, and 4 are applicable for cylinders of any size and for any rate of heat transfer considered in this study, but would require modifications for soil having properties other than those assumed. For example, it is seen from Fig. 4 that in an initial 20-year period about 60 percent of the total heat required would be drawn from the soil at distances greater than 50 ft, and over 20 percent of the total heat required would come from distances greater than 100 ft. It is clear, therefore, that if heat is withdrawn over a long period with no off-setting return of heat, it would not be feasible for several close neighbors each to draw on the earth's heat.

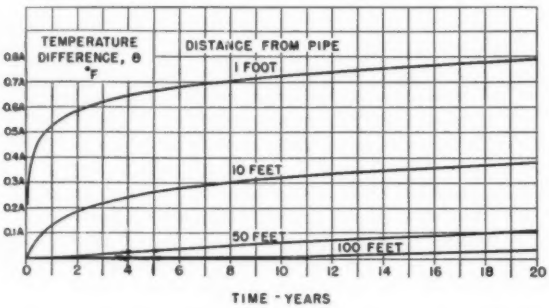

FIG. 2. TEMPERATURE CHANGE FROM INITIAL EARTH TEMPERATURE AT VARIOUS DISTANCES FROM SMALL DIAMETER CYLINDER AS RELATED TO TIME

Heat absorption rate = A Btu/(hr) (linear ft). Earth constants: $k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

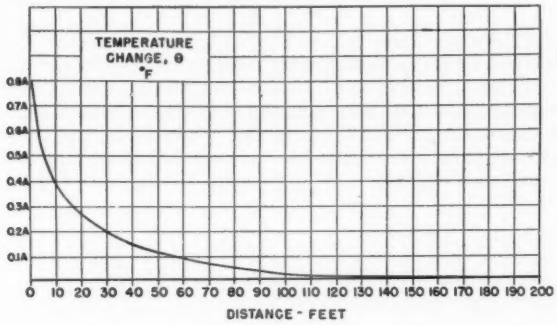

FIG. 3. TEMPERATURE CHANGE FROM INITIAL EARTH TEMPERATURE AROUND A BURIED CYLINDER OF SMALL DIAMETER AFTER 20 YEARS' OPERATION AS RELATED TO DISTANCE FROM CYLINDER

Heat absorption rate = A Btu/(hr) (linear ft). Earth constants: $k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

Likewise the cylinder lengths derived earlier in this paper are to be understood as applying to the case of a single continuous cylinder, as for example, one placed vertically in the earth, that is, the indicated lengths *could not* be changed into two or more shorter lengths unless wide spacings between the separate portions were maintained. This conclusion would be the same with partial return of heat if there were still a substantial unbalance between heat withdrawn and returned. On the other hand, it will be indicated later in this paper that such interference between neighboring units is much less marked if the quantities of heat withdrawn from and returned to earth are equal.

CONSTANT HEAT WITHDRAWAL FROM PLANAR SURFACE

Case 2: Heat is withdrawn at a constant rate from an infinite planar surface in contact with earth of infinite extent in both directions from the surface and at uniform initial temperature. Since edge effects are ignored, the solution of this problem will be more accurate the larger the surface used.

A practical analogue to the case considered is a grid of small pipes so closely spaced that they act in effect as a heat receiving plate. It is not within the scope of this paper to demonstrate the spacing of pipes required. It may, however, be said briefly that for this case of constant rate of withdrawal of heat, a spacing of three or four feet between pipes is the near-equivalent of a plate.

An equation applicable to this case is given by Carslaw and Jaeger[5] as

$$\theta = \frac{B}{2k}\sqrt{\frac{\alpha}{\pi}}\int_0^t e^{-\frac{x^2}{4\alpha t}}\frac{dt}{\sqrt{t}} \quad \ldots\ldots\ldots\ldots (3)$$

For the special case of the planar surface where $x = 0$, Equation 3 becomes

$$\theta_s = \frac{B}{k}\sqrt{\frac{\alpha t}{\pi}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

If the same soil properties are assumed as were used in the calculations relating to the cylindrical heat transfer surfaces, the relation over a 20-year period between temperature at the heat transfer surface and time for any given heat withdrawal rate, B, is as shown in Fig. 5. With the planar surface a much smaller part of the total temperature change occurs in the early years than with cylindrical surfaces (Fig. 1).

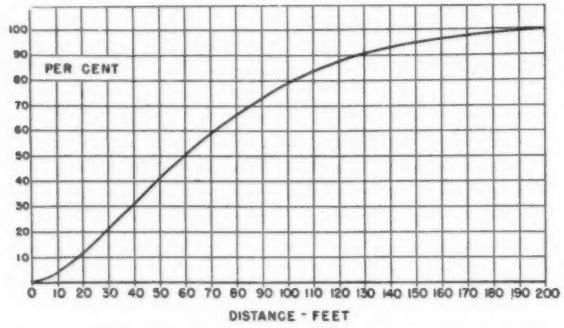

FIG. 4. CUMULATIVE CONTRIBUTIONS TO TOTAL HEAT ABSORPTION OVER A 20-YEAR PERIOD AS RELATED TO DISTANCE FROM CENTER OF SMALL DIAMETER CYLINDER

Heat absorption rate = A Btu/(hr) (linear ft). Earth constants: $k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

If it is again assumed that the temperature of the heat transfer surface may drop at the end of 20 years to a point 40 F below the original soil temperature, it is found (noting that the value of the ordinate of the curve at time = 20 years is 36.5 B), that the rate of heat extraction, B, may not be more than 40/36.5 Btu per hour per square foot of plate. The plate area for continuous heat withdrawal of 50,000,000 Btu per year for 20 years at a constant rate is hence 50,000,000/8760 × 36.5/40 = 5210 sq ft. This is for the case where heat is flowing to the plate from both directions, as when the plate is placed vertically in the earth. The total heat receiving surface would thus be about 10,500 sq ft, under the generally favorable conditions assumed.

The plate size needed to supply heat on an actual schedule of requirement would of course be increased. It is estimated, however, that the increase would be relatively small, perhaps one-fourth. (This factor fits the case where the heat flow is at a rate of 5710 Btu per hour for 20 years, after which heat is withdrawn at the rate of 18,000 Btu per hour for three months.) There is apparently an extensive and available reserve to meet peaks in demand for heat, with only slow drop in temperature in the vicinity of the planar surface. This is not the case with the cylindrical surface.

PERIODIC WITHDRAWAL AND RETURN AT PLANAR SURFACE

Case 3: Heat is periodically withdrawn from and returned to the earth at a planar heat transfer surface as provided by a plate at such a rate that the temperature of the surface varies sinusoidally with time. The amounts of heat extracted and returned are the same. The earth is thus considered essentially as a heat storage medium rather than as a heat-source. If the heat which would be returned to the earth, incidental to summer cooling, is less than the heat required from the earth for winter heating, the deficiency is to be made up, as for example from summer heat.

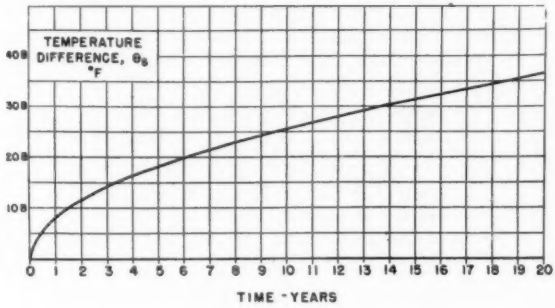

FIG. 5. TEMPERATURE DIFFERENCE BETWEEN SURFACE OF FLAT PLATE AND INITIAL EARTH TEMPERATURE AS RELATED TO TIME

Heat absorption rate $= B$ Btu/(hr) (sq ft). Earth constants: $k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

Here again, the practical analogue of a plate of a given area is a grid of small pipes sufficiently close and covering the same area.

Following the method outlined by Ingersoll and Zobel[6], it may be shown that as the temperature of the planar heat transfer surface, θ_s is varied sinusoidally, (that is, according to the relationship $\theta_s = \theta_o \sin \omega t$) the flow of heat to and from the heat transfer surface is described by the equation:

$$Q = k \sqrt{\frac{\pi}{\alpha T}}\, \theta_o [\sin\omega t + \cos\omega t] \quad \ldots\ldots\ldots\ldots\ldots (5)$$

The imposition of a sinusoidal variation in either heat flow or temperature has thus the effect of causing, after an initial period of a few years, both the temperature of the heat transfer surface and the heat flow to vary sinusoidally, the curve for heat flow leading that for temperature by 1½ months, as shown in Fig. 6.

The assumption of a sinusoidal variation in heat flow to and from a ground coil for heat pumps would appear to be reasonable as a first approximation, in view of the fact that the atmospheric temperature to which heating and cooling requirements are related, follows a sine curve quite closely, as shown in Fig. 7.

Here it is indicated that the average yearly temperature for Baltimore is 56 F, and the amplitude of the temperature variation is 22 F. It is true that the actual schedule of heat flow to and from earth which would be required for the heat pump would deviate in practice from the sine curve. A consideration of the effect of these deviations indicates that the actual cylinder length required should be somewhat greater than for the idealized *Case 3*.

For soil of the same favorable properties as those assumed in the previous examples, Equation 5 becomes

$$Q = 0.122\ \theta_0\ (\sin\omega t + \cos\omega t) \quad \ldots\ldots\ldots\ldots (6)$$

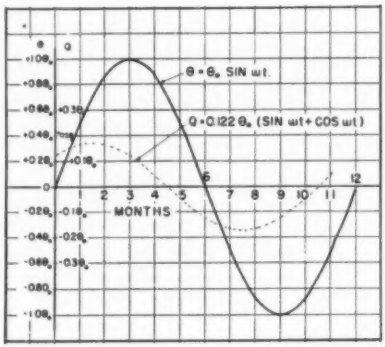

FIG. 6. HEAT FLOW RATE AT THE SURFACE OF A BURIED PLATE FOR A SINUSOIDAL TEMPERATURE VARIATION OF THE PLATE

Earth Constants: $k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

The total heat flow, W, into or out of the earth corresponding to one loop of the sinusoidal heat flow curve (see Fig. 6) is

$$W = 0.122\ \theta_0 \int_{t=\frac{3\pi}{4}}^{t=\frac{7\pi}{4}} (\sin\omega t + \cos\omega t)\, d\omega t \quad \ldots\ldots\ldots (7)$$

For soils having properties other than those assumed here, other constants would be obtained instead of the value 0.122.

If as in other examples it is assumed that a variation from original soil temperature of 40 F is permitted, (that is, $\theta_0 = 40$ F) the value of W is found to be 19,240 Btu per sq ft. For a seasonal heat withdrawal from the earth of 50,000,000 Btu, the indicated minimum requirement of planar heat transfer surface is $50{,}000{,}000/19{,}240 = 2600$ sq ft. If the surface is in the effective form of a plate so disposed that two surface areas are active, the minimum extent of plate required would be one-half, or about 1300 sq ft.

In the foregoing case, heat is restored to the earth during the six-months period following heat withdrawal, while actually the heat would have to be

restored in four summer months in many localities. This could be accomplished with the indicated surface (or plate) area only if it is feasible to allow the maximum temperature rise of the heat transfer surface in summer to be perhaps one-third greater than the assumed permissible maximum temperature drop of 40 F deg in winter. If in the practical case, the temperature departure in summer is limited to 40 F deg, the required surface would be appreciably increased. In such a case, there would be the incidental benefit that the maximum winter drop would be less than 40 F.

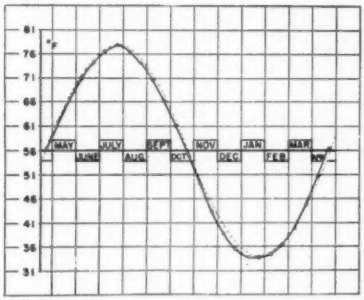

FIG. 7. AVERAGE SEMI-MONTHLY TEMPERATURES FOR BALTIMORE SHOWING CLOSENESS OF SEASONAL VARIATION TO A SINUSOIDAL CURVE

(Points shown are averages for 46 years, 1875-1921, dotted curve is a theoretical sine curve.)

PERIODIC WITHDRAWAL AND RETURN AT CYLINDRICAL SURFACE

Case 4: Heat is periodically withdrawn from and returned to the earth in equal amounts at a cylindrical heat transfer surface at such a rate that the temperature of the surface varies sinusoidally with time. Here, as in *Case 3,* the earth is used as a heat storage medium rather than as a primary heat source.

The fundamental differential equation for heat flow to or from a cylindrical surface in contact with an infinite solid is,[7]

$$\frac{\partial^2\theta}{\partial r^2} + \frac{1}{r} \cdot \frac{\partial\theta}{\partial r} - \frac{1}{\alpha} \cdot \frac{\partial\theta}{\partial t} = 0 \quad \ldots\ldots\ldots\ldots (8)$$

By a derivation the presentation of which is beyond the scope of this paper, but which is adapted from McLachlan,[8] and from Carslaw and Jaeger,[9] taking account also of the condition that $\theta_s = \theta_o \sin \omega t$, it is found that

$$\theta = \theta_o \frac{N_o\left(r\sqrt{\frac{\omega}{\alpha}}\right)}{N_o\left(a\sqrt{\frac{\omega}{\alpha}}\right)} \sin\left[\omega t + \varphi_o\left(r\sqrt{\frac{\omega}{\alpha}}\right) - \varphi_o\left(a\sqrt{\frac{\omega}{\alpha}}\right)\right] \ldots\ldots (9)$$

and

$$q = 2\pi a \theta_0 k \sqrt{\frac{\omega}{\alpha}} \frac{N_1\left(r\sqrt{\frac{\omega}{\alpha}}\right)}{N_0\left(a\sqrt{\frac{\omega}{\alpha}}\right)} \sin\left[\omega t + \varphi_1\left(r\sqrt{\frac{\omega}{\alpha}}\right) - \varphi_0\left(a\sqrt{\frac{\omega}{\alpha}}\right) + \frac{3}{4}\pi\right] . \quad (10)$$

For the special case of heat flow at the surface ($r = a$):

$$q_a = 2\pi\, a\, \theta_0\, k \sqrt{\frac{\omega}{\alpha}} \frac{N_1\left(a\sqrt{\frac{\omega}{\alpha}}\right)}{N_0\left(a\sqrt{\frac{\omega}{\alpha}}\right)} \sin\left[\omega t + \varphi_1\left(a\sqrt{\frac{\omega}{\alpha}}\right) - \varphi_0\left(a\sqrt{\frac{\omega}{\alpha}}\right) + \frac{3}{4}\pi\right] . \quad (11)$$

Values of the functions N_0, ϕ_0, N_1, ϕ_1, are given by McLachlan and Meyers[10] together with expansion formulas for calculating values not tabulated. The values needed for calculations relating to the surface of pipes of three different diameters and for soil of the properties assumed in previous examples are:

DIAMETER	a	$s = a\sqrt{\frac{\omega}{\alpha}}$	N_0 (s)	ϕ_0 (s)	N_1 (s)	ϕ_1 (s)
¾ in.	0.0313	0.006	5.182	− 8.72°	166.66	224.99°
2 in.	0.0833	0.016	4.323	−10.47°	62.49	224.96°
3.82 in.	0.1592	0.030	3.707	−12.22°	33.32	224.89°

Substituting these values in Equation 11, the following equations for heat flow rates applicable for soil of the assumed properties are obtained:

For the ¾ in. pipe $q = 1.09\ \theta_0 \sin(\omega' t + 8.71)^\circ$ . . . . . (12)
For the 2 in. pipe $q = 1.31\ \theta_0 \sin(\omega' t + 10.43)^\circ$ . . . . . (13)
For the 3.82 in. pipe $q = 1.57\ \theta_0 \sin(\omega' t + 12.11)^\circ$ . . . . . (14)

where ω' is the angular velocity in degrees per hour.

These equations are plotted as the dotted curves in Fig. 8, and show the way in which heat flow varies as the temperature is varied according to the solid curve. The phase displacement between the temperature and heat flow curves is from 0.3 to 0.4 months as compared with 1.5 months for the planar heat transfer surface (Fig. 6), and is the less the smaller the cylinder.

The total heat flow into or out of the earth in each season corresponding to one loop of the sinusoidal heat flow curve may be obtained by integrating Equations 12, 13, and 14 which become:

For the ¾ in. pipe

$$W' = 1.09\ \theta_0 \int_{\omega' t\,=\,171.29^\circ}^{\omega' t\,=\,351.29^\circ} \sin(\omega' t + 8.71)^\circ\, d(\omega' t) = 3{,}040\ \theta_0 \text{ Btu/ft} \quad . \quad . \quad . \quad . \quad (15)$$

For the 2 in. pipe

$$W' = 1.31\ \theta_0 \int_{\omega' t\,=\,169.57^\circ}^{\omega' t\,=\,349.57^\circ} \sin(\omega' t + 10.43)^\circ\, d(\omega' t) = 3{,}665\ \theta_0 \text{ Btu/ft} \quad . \quad . \quad . \quad . \quad (16)$$

For the 3.82 in. pipe

$$W' = 1.57\ \theta_0 \int_{\omega' t\,=\,167.89°}^{\omega' t\,=\,347.89°} \sin\,(\omega' t + 12.11)°\ \mathrm{d}(\omega' t) = 4{,}380\ \theta_0\ \text{Btu/ft} \quad . \quad . \quad . \quad . \quad (17)$$

where W' is total heat withdrawn or returned in one year per foot of cylinder length.

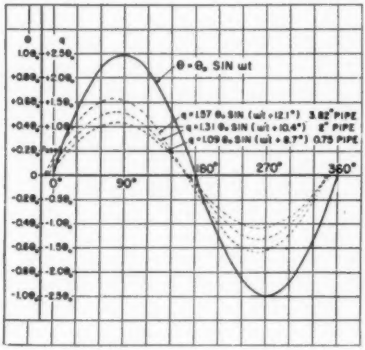

FIG. 8. VARIATION OF HEAT FLOW RATE AT THE SURFACE OF BURIED PIPES OF DIFFERENT DIAMETERS WHEN THE TEMPERATURE AT THE PIPE SURFACE VARIES SINUSOIDALLY

Earth Constants: $k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

These results are summarized as follows:

Cylinder Diameter—In.	¾	2	3.82
Seasonal Heat Flow—W'—Btu/Ft	$3040\theta_0$	$3665\theta_0$	$4380\theta_0$
Seasonal Heat Flow for $\theta_0 = 40$, W'_{40}—Btu/Ft	121,600	146,600	174,200

If now, as in the preceding examples, it is assumed that a temperature variation from original earth temperature of 40 F is to be permitted, (that is, $\theta_0 = 40$) and that the total heat withdrawn (or returned) in one season is 50,000,000 Btu, then the indicated lengths of cylinder, L, required are, $50{,}000{,}000/W'_{40}$, or:

For ¾ in. pipe $L = 411$ ft
For 2 in. pipe $L = 342$ ft
For 3.82 in. pipe $L = 286$ ft

The cyclic temperature variations in the soil resulting from the imposed heat flow at the cylindrical surface may be calculated from Equation 9. The amplitude, or maximum difference from original soil temperature, of these variations

will decrease rapidly with increasing distance from the cylinder as shown in Fig. 9, the curves being those calculated for soil of the assumed properties. It will thus be seen that for the case of cyclic withdrawal and return of heat, as contrasted with withdrawal only, separate cylinders may be located within short distances from each other, say 20 or 30 ft, without substantial interference.

DISCUSSION

When the length of an isolated straight pipe is considered for withdrawal only of heat over a 20-year period on a schedule of *actual* requirements for normal heating, it is clear that the length would in any case be greater than that required for balanced yearly withdrawal and return of heat. It is estimated that the length would be 25 to 50 percent greater. A new estimate of 400 to 500 ft, depending on the pipe diameter, is therefore reached for pipe length for heat withdrawal without return. This is $2\frac{1}{2}$ to 3 times the preliminary estimates of length based on heat withdrawal at a uniform rate. The corresponding factor in the case of a plate (or equivalent grid) is much smaller, being about $1\frac{1}{4}$ as noted previously.

A re-examination of the calculations relating to continuous withdrawal of heat at a plate surface, *Case 2*, is necessary to determine limitations in their practical application. It should be recalled that effects of heat exchange with atmosphere were excluded in those calculations, and hence the results apply strictly to the case of a deeply buried plate (or equivalent pipe-grid).

Such an arrangement would hardly be considered in practice, however, unless the extent of surface required would be less for deeply buried coils than for the more convenient shallow coils. It can be shown that, under selected operating conditions, with coils having relatively shallow ground cover, much or even most of the heat extracted by the coil would actually be drawn from the

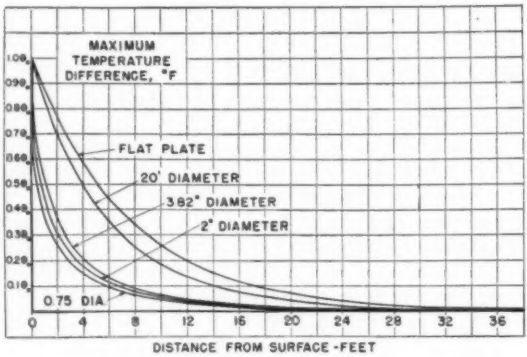

FIG. 9. MAXIMUM VARIATION FROM BASE TEMPERATURE AT POINTS IN THE EARTH REMOVED FROM PIPE

For sinusoidal variation at the surface and earth constants:
$k = 0.896$, $c_p = 0.45$, $\rho = 103$, $\alpha = 0.0193$

TABLE OF SYMBOLS

θ = temperature change with respect to the original uniform earth temperature, Fahrenheit degrees, a variable with respect to time and distance.
θ_s = temperature change at the surface, Fahrenheit degrees.
θ_o = amplitude of temperature variation, Fahrenheit degrees.
k = mean thermal conductivity of soil, Btu per (square foot) (hour) (Fahrenheit degree per foot).
c_p = mean specific heat of soil, Btu per pound per Fahrenheit degree, a constant over the temperature range involved.
ρ = density of soil, pounds per cubic foot, a constant.
$\alpha = k/\rho c_p$ = thermal diffusivity of soil, square feet per hour, a constant.
q = rate of heat flow, Btu per (hour) (linear foot of cylinder).
A = imposed constant rate of heat flow at the cylindrical surface, Btu per (foot of axial length) (hour).
Q = rate of heat flow, Btu per (hour) (square foot of planar heat transfer surface).
B = imposed constant rate of heat flow at the planar surface, Btu per (square foot) (hour).
W = total heat, Btu per (season) (square foot of planar heat transfer surface).
W' = total heat, Btu per (season) (linear foot of heat transfer cylinder).
T = period of complete time cycle, hours (1 year = 8760 hours).
t = time, hours.
$\omega = 2\pi/T$, radians per hour.
$\omega' = 360/T$ degrees per hour.
a = radius of cylinder, feet.
r = radial distance from the axis of the cylinder to a point in the soil, feet, a variable.
x = distance perpendicular to the heat transfer surface, feet.
$k = 0.896$.
$c_p = 0.45$.
$\rho = 103$.
$\alpha = 0.0193$.

atmosphere rather than from the earth. The extent of such coils may be somewhat less than the 13,000 sq ft calculated for the conditions of *Case 2*. A consideration of shallow coils at various depths suggests, however, that not less than 6000 to 7000 sq ft or overall coil extent would be required to supply the heat requirements of the small house considered, if the minimum temperature at the evaporator is to be not lower than 15 to 20 F, as implicitly assumed in previous examples. The required surface would drop rapidly with a reduction in minimum permissible temperature at the evaporator.

It has been indicated that when the heat transfer surface is in the form of a plate (or equivalent planar coil) the surface required is only about one fourth as great for balanced heat extraction and return (*Case 3*) as for heat withdrawal only (*Case 2*). The relatively small surface for the cyclic case would, however, be obtainable in practice only if heat interchange with the atmosphere were minimized. This would require that the heat transfer surface be located deep in the earth. The coil might, for example, take the form of a row of vertical pipes driven into the earth at sufficiently close intervals to act effectively as a plate.

While discussions of this paper have been concerned with the movement of heat through the soil by conduction only, consideration has been given to other

possible factors affecting heat transfer. Such factors as thermal migration and convection of liquid water and diffusion of water vapor were considered in sufficient degree to suggest that conduction is the dominant factor in heat transfer, except in those special locations where there is a substantial gravity flow of ground water near the heat transfer surfaces.

Considering the generally favoring bias of the assumptions in this study, it is believed that in most cases ground coils would have to be as large as or larger than indicated. Since, however, even as estimated here, the extent of coil needed for the various cases is quite large from the standpoint of installation cost, it is doubted that ground coils will be an economic heat-source for *general use.* It appears, therefore, that as much attention as possible should be given to the study of means of minimizing the difficulties in the use of other sources, especially air.

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DISCUSSION

C. W. BROWN, Bloomfield, N. J. (WRITTEN): I agree with the authors' conclusion that as much attention as possible should be given to the study of means of minimizing the difficulties in the use of other heat sources, especially air.

My reasons for agreement are largely independent of the reasons given by the authors and are based on the sometimes unanswerable problems that face the application engineer considering the ground as a heat source. He encounters the following questions:

1. What is the probable heat transfer rate of the soil?
2. Is there rock strata that will interfere with placement of a ground coil?
3. Will too much sales resistance be created if he asks the owner to permit digging up his lawn, flower garden, drive or placement of the ground coil under the foundation?

4. Does the customer own sufficient land area to permit placement of a ground coil large enough to heat the building?
5. How much will it cost to detect and correct leaks in the ground coil?

Since the application engineer considering air as a heat source faces none of these questions, he may logically prefer, and use his influence, to swing the job to air as a heat source. He might prefer air as a heat source because:

1. Heat transfer rates are known when air is used as a heat source.
2. Air is universally available.
3. No sales resistance is created through the use of air.
4. Leaks are in visible locations and detected and corrected by means familiar to the refrigeration industry.

It is significant that at the last meeting of the A.S.H.V.E. Technical Advisory Committee on the Heat Pump members voted to emphasize heat source investigation in the following order: (1) air; (2) earth; (3) solar; (4) ground water; and (5) surface water.

W. A. HADLEY*, New York, N. Y. (WRITTEN): The authors are to be congratulated on their paper which presents some very interesting analytical data and the Lord Kelvin line sink theory in a more easily used form for ground coils than has hitherto been available. The Heat Pump Committee of the *Edison Electric Institute* has had several occasions to use line sink theory calculations, and up until now they have been quite laborious.

In connection with this work it is my experience that line sink theory predictions do not correlate in the majority of cases with actual test results. In five experimental installations studied by me, the percent deviations of the temperature differences were as follows: +51, +82 with unfrozen soils, and −11, +101, +206 with frozen soils. Consideration of these findings would indicate that in special cases the line sink theory calculations may be useful, but in general they should be used with caution.

It is interesting to note that in all but one case the temperature difference predicted by line sink theory was significantly larger than the temperature difference actually required. This would mean that ground coils sized by line sink theory would be correspondingly oversize.

The reasons for the given discrepancies have not been adequately explored. Professor Coogan** indicates that it is associated with the frozen soil. There is also evidence† indicating that it is associated with the moisture in the soil and its movement.

The authors state that they considered such factors as thermal migration, and convection of liquid water and diffusion of water vapor in sufficient degree to indicate that simple conduction is the dominant factor in heat transfer in soils. It would be very valuable if the authors would give the details of the studies they made in arriving at this conclusion as there is little information available on this subject in the literature at this time.

R. C. JORDAN, Minneapolis, Minn. (WRITTEN): The authors have presented a very interesting analysis of the limitations under which ground coils, when used as a heat source, are called upon to operate. They correctly indicate them as economically questionable when applied under the conditions outlined in the paper. They however purposely exclude migration and freezing of liquid water, diffusion of water vapor, and consideration of heat supplied from the atmosphere rather than the earth.

In Cases 3 and 4 of the solutions presented, the ground is considered essentially as a medium for heat storage rather than as a heat source in itself. This concept is proper as the actual source of much of the heat is, either directly or indirectly, solar.

* Assistant professor of mechanical engineering, Columbia University.

** *Heat Transfer Rates*, by C. H. Coogan (*Mechanical Engineering*, Vol. 71, No. 6, June 1949).

† *The Effect of Soil Moisture on Buried Receivers for Heat Pumps*, by W. A. Hadley (Electrical Engineering Bulletin, Vol. 17, No. 1, January 1949).

Thus the problem of the correct and economical application of ground coils may prove eventually to be one of design for maximum solar energy recovery when the earth is being used as a heat source and not a heat sink. This indicates the need for further research into the effects of various surface and sub-surface soil treatments, the application of solar concentrators, and means for the utilization of the latent heat of fusion of different materials, including water.

There has never been any question as to the vast amounts of solar energy available. The problem has been to capture reasonable portions of this heat without utilizing exceedingly bulky, ungainly, and hence impractical equipment. Since it is more efficient to absorb this solar heat at the comparatively low temperatures practicable for the heat pump heat sources than at the higher temperatures attempted for other purposes in the past, this holds some promise. Whether this can actually be done remains for more complete investigation, but all developments in this direction will point to a more promising future for ground coils than the dubious one indicated by the results of the present researches.

C. F. KAYAN, New York, N. Y. (WRITTEN): The authors, in making their most welcome contribution, do well to advocate caution in regard to the general possibility of using the earth as a *reverse cycle* heat source. This is most important lest uninformed enthusiasts rush in with ground coil installations where climatic conditions are severe and heating requirements predominating, only to achieve discouraging and discrediting failure. Direct heat sources such as air offer more reliable basis for safe prediction of performance.

Aside from the obvious mathematical difficulties due to non-uniformity of physical properties of earth and due to variation in initial temperature conditions in the region affected by the heat absorbing surface, the applicability of some of the available mathematically derived equations to this present specific problem must continue to be scrutinized critically. For example, would it be reasonable to assume that a vertical equivalent flat plate one foot wide, installed vertically to an extent of 5,000 ft, would behave the same as an equivalent vertical flat plate 50 ft high and 100 ft wide, or as another one 10 ft high and 500 ft long? And should the performance of the one-foot wide plate be closer to that of the separate pipes, such as the 3.82-in. tube, than as at present indicated?

It is reasonable to assume that a grid of small closely spaced pipes will act in effect as a continuous receiving plate. Verification for this is indicated by electric Analogger* panel heat flow studies by the writer. It was found that with imbedded pipes under steady-state conditions and with decreasing heat flux densities, the panel more and more tends to reach toward uniform temperature conditions.

J. D. KROEKER, Portland, Ore. (WRITTEN): An Oregonian has undoubtedly been requested to comment on this paper because residential heat pumps in Oregon using earth as a heat source have received national publicity through news services and coast-to-coast radio hook-ups, informing the world that these applications have been failures.

There were some mis-applications following the postwar heat pump hysteria. Most of the Oregon firms which sprang up overnight to fill the demand—and unfortunately to promote it, too—are now bankrupt. A number of installations have been converted to oil. Yet there are also more than a simple majority of these installations which operated satisfactorily throughout the 1948-49 winter, Oregon's longest and coldest heating season.

This paper presents an analysis which is a very logical presentation of known mathematical formula and soil conductivities. However, some conclusions seem open to question, on the basis of observations in installations using earth as a heat source.

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* Heat Transfer and Temperature Distribution in a Panel with Imbedded Pipe, by C. F. Kayan (*Refrigerating Engineering*, August 1947).

Some of the difficulty in checking the figures derived mathematically against the results obtained in practice, can lie in the fact that the mathematical analyses are based upon conductivities, while the results obtained in practice depend to a large extent on transfer factors other than conductivity. Whether it is conduction or convection which actually transfers the heat from one point to another in the ground may be open to discussion. The fact remains that, in many soils with a 20 to 30 percent moisture content, a heat transfer rate of 2 to 10 Btu per (sq ft) (hr) (F deg) is often obtained.

Almost every engineer dealing with ground source heat pumps has gone through conductivity calculations to determine the amount of surface required and obtained an answer which made use of a ground source for the heat pump impracticable. Yet, there are many heat pumps operating today with ground coils having surfaces a small fraction of those indicated from calculations. It is also true that some of the ground coils do not operate satisfactorily. Where the coils have been unsatisfactory one or both of two conditions were found to exist. *First,* an absence of moisture or moisture movement in the soil, due either to the impervious nature of the soil or due to dry weather; *second,* concentration of the coil in a limited volume of soil such that the ground froze and expanded to such an extent as to cause extensive damage. In all cases the amount of ground coil used was only a fraction (generally, 15 to 30 percent) of the amount indicated as required by conductivity calculations.

A third factor which has caused trouble with ground coils in Oregon has been their location too far below the surface of the earth. A possible explanation of this phenomenon is the observed fact that, where average winter temperatures are in the neighborhood of 40 F to 45 F and an evaporator temperature of 20 F is maintained, the temperature of ground will increase 5 to 10 deg in a few days of rainy weather after a cold dry spell during which the entire area covered by the ground coil has dropped to freezing temperatures.

There is still much to be learned about ground coils, but experience with them to date indicates that, in soil which allows free movement of moisture, both liquid and vapor, located in areas with mild, wet winters, the ground coil can operate successfully and be reasonably economical to install. This is true even where the soil is not used as a heat pump. Studies of heat pumps actually using ground coils are needed. Many data are being assembled on such installations and should be coordinated. Oregon Chapter A.S.H.V.E. has in progress its own unsubsidized, fully-instrumented research project on one such ground coil installation on which data from a full year of operation should be available next year.

Finally, the paper is a sound analysis only of conductivity of earth but not an analysis of practical application of a coil located in earth to withdraw heat therefrom. It will serve, too, as a means of attracting the designer's attention to the very real limitations which must be considered in using dry earth as a heat source.

E. P. PALMATIER, Syracuse, N. Y. (WRITTEN): The authors of this paper have presented in convenient form the following new material. *First,* they have shown by Fig. 4 and the pertinent discussion that a relatively large percentage of the total heat removed from the ground by a refrigerated pipe comes from considerable distance if there is no heat return to the earth. *Second,* they have presented the equations and by means of Fig. 6 have shown graphically the manner in which the heat flow to a plane refrigerated surface varies if the temperature of the surface is varied sinusoidally both above and below the undisturbed ground temperature which is assumed to be constant. *Finally,* in Case 4 and Figs. 8 and 9, the authors have shown the relationship between heat withdrawal and surface temperature for a single cylinder in an infinite mass of earth when the surface temperature is varied sinusoidally above and below the undisturbed ground temperature. Fig. 9 emphasizes the fact that when heat is withdrawn and returned in equal amounts the temperature of surrounding earth

remains relatively unaffected at distances much closer than in the case of continuous heat removal.

These items are definite contributions to the theory of heat pump heat exchangers buried in the earth. The authors are to be congratulated for presenting the material in such clear and usable form. The principal criticism I have of the paper is that the authors have given the impression that the several cases presented are examples of the actual design of earth heat exchangers for a residential heat pump and that the problem is solved, and it may be concluded that a ground coil is economically impractical. Early in the paper they lay down a set of assumptions which they say corresponds to the case of a small, well-insulated house in an area where the minimum temperature is 0 F and the temperature deficiency is 4500 degree days per season, etc. In the very next paragraph, it is generally assumed for the calculations that the temperature of the heat transfer surface may deviate from the normal average earth temperature as much as 40 deg. It would seem likely that the authors had in mind their home city which has 4487 degree days, compared to the 4500 assumed, and a minimum design temperature of 0 F as assumed. According to water supply paper #520-F published by the *U. S. Geological Survey*, the ground water temperature in Baltimore is 57 F. Using this temperature as representative of the average ground temperature and the figure of 40 deg for the maximum temperature difference between undisturbed ground and surface, a surface temperature of 17 F is obtained.

With a surface temperature of 17 F moisture will be frozen in the soil and under these conditions, I submit that the latent heat contributed by this freezing requires the use of an entirely different theory. Let me present here just a few figures to indicate why I believe some other theory other than that involving simple heat capacity and conduction must be used if moisture is frozen in the soil around a heat pump coil.

The following material is extracted from unpublished studies conducted by me about two years ago. Let us assume a fairly moist sandy soil having the following properties:

Density, wet	= 132 lb per cu ft
Moisture, dry basis	= 20 percent
Moisture content	= 22 lb per cu ft
Specific heat	= 0.25 Btu per (lb) (F deg)

Let us now assume that soil of the above properties can be progressively frozen to a diameter several feet around a buried tube, the surface temperature of which is 20 F. The soil at the outer limit of freezing will be close to 32 F. The temperature gradient in the frozen soil will be essentially logarithmic with respect to radius if thermal capacity in the frozen soil is of secondary influence. Then the mean temperature of the frozen soil will be nearer to 32 F than 20 F. Let us assume a mean temperature of 29 F and an initial temperature of 50 F and compare the relative values of the latent heat of fusion and the sensible heat which is extracted from the frozen soil.

For each cubic foot the sensible heat corresponding to the density of 132 lb per cu ft, the specific heat of 0.25 and the temperature depression of 21 deg is 694 Btu per cu ft. The latent heat of fusion of the 22 lb of moisture is 3,170 Btu per cu ft. The total is 3864 Btu per cu ft of which the latent heat of fusion accounts for 82 percent, whereas the sensible heat accounts for only 18 percent. Of course while the freezing is in progress, an additional quantity of heat will be absorbed by the tube. This heat will have been drawn from outside the volume of earth which is eventually frozen. However, as the authors of this paper have indicated, this conducted heat must come from a considerable distance from the tube and consequently, if adjacent tubes are present, their interferences reduce the quantity of heat which can be obtained from outside the immediate vicinity of each tube.

Fig. A shows the relationship between radius of frozen soil, loading per foot of tubing and hours of operation for a ¾ in. pipe buried in soil containing 22 lb per

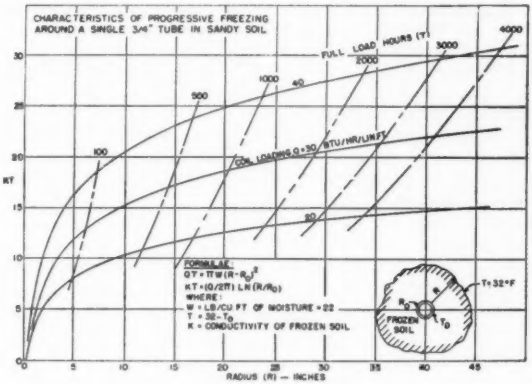

FIG. A. CHARACTERISTICS OF PROGRESSIVE FREEZING AROUND A SINGLE ¾ IN. TUBE IN SANDY SOIL

cu ft of moisture, *neglecting entirely the sensible heat of the soil and any conducted heat which may come from the surrounding soil.* In other words, this diagram embraces only the latent heat of fusion of the soil moisture and steady state conduction from the freezing layer inward to the tube. The ordinate is the product of conductivity of the frozen soil and depression of the tube surface temperature below 32 F.

Now if we use the authors' figures of 4500 degree days and 75,000,000 total Btu per season we find that the house assumed has a maximum heat loss of 45,000 Btu per hr, which means that 30,000 Btu per hr must be extracted from the ground. By means of Fig. A with a loading of 35 Btu per linear ft of tube, we find that when using 860 lineal feet of tube, the ice will have frozen to a radius of 28.7 in. after an equivalent of 1670 full load operating hours with a total heat removal of 50,000,000 Btu. The corresponding value of KT is 24. To find the depression of tube surface temperature below 32 F we must know the conductivity of the frozen soil, K. Professor Coogan has reported experimental values of conductivity for frozen soil in the vicinity of 1.5. If such a value is used for the conductivity, we arrive at a temperature difference between tube surface and 32 F of 16 deg.

It seems evident, therefore, that if soil containing considerable moisture can be frozen, an adequate source of heat is available, at least for one heating without going more than 2 or 3 ft from the buried tube. We may, therefore, begin searching for means of reducing the temperature difference between 32 F and the surface temperature of the tube. When my analysis was conducted, it seemed that the cost of trenching for horizontal ground coils would overshadow the cost of the tube material so the case of two parallel tubes in the trench was studied. This Fig. B shows the effect of two tubes placed 6 in. apart. The ordinate in this case includes a new variable, M, which is the *strength* of the *thermal sink* or the number of Btu per (hour) (lineal foot per radian) absorbed by each of the two tubes. On this same chart are shown (1) the general relationship for a single tube in which case the abscissa is the ratio of radius of the frozen cylinder to the radius of the tube (applicable to relatively small tubes only) and (2) the relation for a single ¾ in. tube in which case the abscissa is the radius of the frozen cylinder in inches.

One of the reasons why these data were never published was that I was not too sure of the line sink theory as applied to this problem. You will notice that the curve

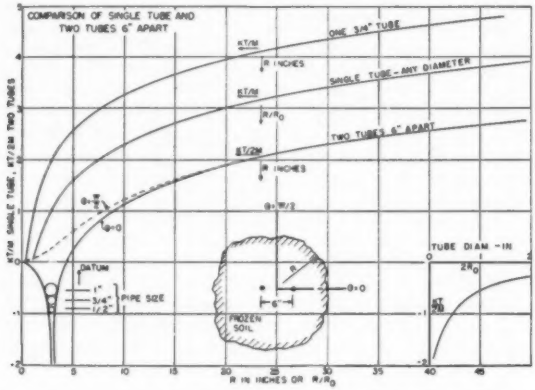

FIG. B. COMPARISON OF SINGLE TUBE AND TWO TUBES 6 IN. APART

for the case in which two tubes are buried in the trench has a cusp which goes to minus infinity at the location of the tube. This is so because a line sink is assumed at this location and consequently the temperature gradient becomes infinity as one approaches the point corresponding to the tube location. In my work I assumed that the datum for any particular tube diameter would be at that value of $KT/2M$ where the horizontal space between the two curves of the cusp is equal to the tube diameter. In the lower righthand side of Fig. B is a curve showing the relationship between tube diameter and the depression of the datum.

These data would indicate that the single tube and two-tube curves are essentially parallel for values of R above about 10 in. Consequently, the percentage improvement is greater for smaller diameters of the frozen cylinder. For maximum diameters of the frozen cylinder of 2 to 3 ft which would probably be the order of magnitude for most economical design, the temperature depression of the tube surface below 32 F appears to be about 40 percent greater for a single tube than for two tubes of the same diameter. This comparison applies of course for the same loading per linear foot of trench. The loading for each of the two tubes is only half as great as that of the single tube.

Returning now to the problem we find that for two tubes and a loading of 35 Btu per hr per linear ft of trench, we arrive at a value of KT of 16.7 or a temperature differential of about 11 deg instead of 16 deg which was obtained for the single tube.

In Case 4 the authors obtained a length of 411 ft for a ¾ in. pipe with cyclic withdrawal and return of heat. Using the soil freezing theory which I have described with a length of trench of 411 ft and two ¾ in. tubes in the trench, I obtained a surface temperature of 6 F or a temperature depression of 26 deg below 32. Conductivity of the frozen soil was taken at 1.5 as before. Obviously this is a lower surface temperature than would be obtained with a 40 deg temperature difference and an undisturbed ground temperature of 57 F. However, the pipes could theoretically be installed horizontally within about 6 ft of each other without serious interferences because the radius of the frozen soil would be approximately 42 in. The conduction theory would indicate that the two tubes would have to be placed vertically with a center distance approaching 20 ft to avoid interference.

To summarize, there are two fundamental approaches to the design of a ground coil for a heat pump:

1. The *conduction theory*, which recognizes the thermal capacity of the ground and which has been treated by the authors of this paper and others. Adequate treatment of this theory for horizontal ground coils involves recognition of the existence of the ground surface above the coils and the normal seasonal temperature variations of this surface. It also involves treatment of the influence of periodic heat removal and return, although it is very doubtful whether the theory can adequately cover such effects as solar radiation on the ground surface during the summer months and the variation in conductivity of the soil caused by the tendency of the coil to dry out the soil surrounding it when heat is being returned (the latter has been observed by several experimenters). A further complicating factor would be the effect of precipitation and the filtering of water downward through the ground from melting snow and rain.

2. The *freezing moisture theory* in the soil would seem to be much simpler mathematically. It indicates that adequate heat should be available, at least for one year's operation, within a few feet of the refrigerated tube if sufficient moisture is present in the soil. However, the simple freezing theory which I have described certainly is not realistic because it does not recognize the additional heat which is collected by conduction nor does it take into consideration the effect of solar radiation on the surface of the ground or the flow of water from precipitation downward past the coil. As suggested in a publication* by Professor Coogan, the most economical horizontal ground coil design will probably always result in some freezing of moisture during periods of peak heating load.

Therefore, before engineers can proceed with confidence in the design of horizontal ground coils for heat pumps some more composite theory of design must be developed to include both the thermal capacity and heat conduction of unfrozen soil and the freezing of moisture in the soil. Furthermore, it would seem that this theory must be correlated and checked by the results obtained from quite a few full scale experimental installations.

J. H. CARTER, St. Louis, Mo.: As a simplification of the storage problem, consider a reservoir of water having a flywheel effect to store up enough summer heat to last through the winter, and vice versa. The reservoir could consist of a buried tank filled with refrigerating coils, building up on ice formation in winter, and melting this ice in summer.

For a house of well-insulated construction, the volume of the tank must be about equal to the volume of the house. This may be all right in theory, but it does not seem entirely practicable or economical.

AUTHORS' CLOSURE: As a background for a reply to the several comments, it appears desirable to recall the authors' point of departure in their work which may perhaps have been insufficiently stressed in the paper. At the time the work was undertaken frequent references to the availability of earth as a ***high temperature*** heat source for the heat pump were appearing in the technical and popular press. A main purpose of the work reported in the paper was to appraise the prospect for the development of such a high temperature source in earth. The conclusion was that it is doubtful that earth will provide such a source economically for *general application.* There appears to be no reason to modify this conclusion even though, as indicated by certain of the comments, heat can be, and is, collected by ground coils in particular situations.

In view of the requirement of general applicability, it is of secondary interest to consider the ways in which the presence of moisture may affect the movement of heat or the availability of latent heat since, as indicated in Kroeker's remarks, moisture

* The Residential Heat Pump in England, by Prof. C. H. Coogan, Jr. (See Fig. 49).

may be unavailable in some locations or under unfavorable weather conditions. In any case, the presence of a high moisture content was implied by the high soil conductivity assumed and to that extent the calculated coil lengths are biased in a direction of giving values which are too low. Similarly, the assumption of a coil surface temperature as low as 40 F deg below the initial earth temperature was intended to be extreme in the light of the consideration of earth as a relatively high temperature source. The purpose was again to give a conservative bias to the conclusions drawn. It was not intended to suggest that actual operation at these below-freezing temperatures was necessarily contemplated as Palmatier very reasonably inferred.

Although factors other than soil conductivity, specific heat and density were neglected, qualitative attention was given, as indicated in the paper, to the probable importance of some other influences, because of the general interest in "ground coils" as distinguished from the use of earth as a primary heat source.

Considerable thought had been given, for instance, to the feasibility of utilizing the latent heat of freezing of water. This is discussed by Palmatier. There is, of course, no question that if a ground coil is operated below 32 F in moist soil, and if the mechanical difficulties resulting from the soil movement on freezing referred to by Kroeker can be surmounted, latent heat can supply a substantial or even a major part of the heat. A precise mathematical treatment of the flow for this case involving both latent and sensible heats would be very useful. Lacking such a formulation, it was concluded through qualitative considerations that for the case of withdrawal only (no return) of heat through a cylindrical surface, the length of cylinder required for a given heat supply would not be substantially reduced by the freezing of moisture below that calculated in the paper. The reason is that, even though upon ice formation the principle source of heat is closer to the receiving surface than without ice formation, the resistance to heat flow, after freezing has proceeded a moderate distance into the soil, is not greatly different than in the absence of freezing. This is so because the resistance is largely concentrated in the paths near the cylinder. The length of cylinder for the moisture-freezing condition might be expected to be reduced moderately, but primarily because of the slight increase in conductivity of the frozen core surrounding the coil as compared with unfrozen soil of the characteristics assumed. It is noted in this connection that Palmatier's calculations do not appear to suggest that the required length of cylinder would be significantly reduced through the occurrence of freezing. It is clear, however, as noted by Palmatier that, with freezing, a closer spacing of cylinders would be possible. It is further thought that freezing may be more important in determining the required extent of coil, under some conditions, as when heat is alternately withdrawn from and returned to the coil in approximate balance. The effects of the freezing of moisture merit further detailed study.

Kroeker infers that the movement of heat in soil surrounding ground coils may depend "to a large extent on transfer factors other than conductivity," and Hadley indicates an interest in the basis for the authors' conclusion that certain factors other than conduction are of minor importance. It is not within the scope of this discussion to present these considerations in detail but they may be discussed briefly. As to diffusion, it is possible to make a straightforward calculation of diffusion rates from well-established constants assuming a concentration gradient of water vapor in air corresponding to saturation at temperatures to be expected at various points about an operating ground coil. If the vapor thus transported is assumed to be condensed at the ground coil, the latent heat contributed is found to be only a few percent of the heat flowing by conduction under the same gradient. Moreover, it does not appear that convection currents either of moisture-saturated air or of liquid water can be expected generally to contribute importantly to the transfer of heat to the coil. This possibility is discounted because it would presuppose an interconnected pore system of large extent. Published geological studies indicate that convection in porous earth

strata is extraordinarily slow[11]. Thermal migration, if by this is meant the movement of liquid water in capillaries under the influence of a temperature gradient as studied by Smith[12], provides no mechanism for continuously moving liquid water through the soil and hence no means for carrying sensible heat to the ground coil. Such capillary movement of moisture under the influence of a temperature gradient occurs only in soils which are less than saturated and is a temporary effect unless by some means moisture is continuously removed from the under-saturated soil. Such means are not afforded by a ground coil.

To discount in turn these named effects as probable substantial contributors to heat transfer does not, of course, preclude the possibility that by some other unnamed or unrecognized effect the presence of water in soil may greatly facilitate the movement of heat and thus account for the unpredictably high heat flow reported by Kroeker, Hadley and others. The course of seasonal temperature changes below the earth's surface have, however, been calculated[13] with fair accuracy from data on thermal diffusivity of the soil alone, disregarding even the effects of rainfall and surface freezing. This fact may be taken as further reason to question the importance of factors other than conductivity and diffusivity under usual soil conditions.

It is impossible because of the lack of data to interpret reported cases of abnormally high heat flows to ground coils referred to by Hadley and by Kroeker or mentioned in the literature. It appears in some instances at least that comparison has been made of observed heat absorption by shallow ground coils with calculations relating to the extraction of heat from earth of infinite extent at average earth crust temperature. Such a comparison is, of course, not valid and it is possible that some of the apparent discrepancies may thus be accounted for. As pointed out in the paper a shallow coil may be designed to absorb much more heat than a deeply buried coil at the same temperature if the temperature is lowered sufficiently. It should be noted that under this condition heat entering the soil from the atmosphere currently or during a recent prior period is the source of most of the heat absorbed. It is obvious, however, that when heat flow is thus increased by lowering coil temperature the goal of a *high temperature* heat source which was once supposed to be a primary reason for developing earth as a heat source has been abandoned.

The remarks of the previous paragraph may be qualified to the extent that they make no direct reference to the possibilities of solar energy as a heat source for the heat pump as mentioned by Jordan. It is, of course, physically possible by special devices to obtain from the action of the sun substantially higher temperatures in the soil or other medium than normally prevail in the winter. Much work has been done on such devices in various places aiming usually, however, at temperatures high enough for direct use for space heating. The problem is one of accomplishing this increased absorption of solar radiation at acceptable cost. It is frequently assumed that the solar heat absorbed by the soil during a particular summer is available for heat supply to ground coils during the following winter. It appears, however, that such a carryover would be quite minor for coils having no more than a few feet of earth cover.

As noted by Hadley, calculations of temperature changes in soil resulting from the continuous withdrawal of heat along a line are greatly facilitated by the use of an empirical equation such as Equation 2. It appears useful to add two additional equations which follow. Work following the completion of the paper has shown the possibility of representing temperature changes at the pipe surface resulting from the operation of heat sources of forms more nearly representative of actual practice than

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[11] Convection Currents in a Porous Medium, by C. W. Horton and F. T Rogers, Jr. (*Journal of Applied Physics*, 16, 367, June 1945).

[12] Thermal Transfer of Moisture in Soils, by W. O. Smith (*Transactions American Geophysical Union*, 511, 1943).

[13] Heat Conduction, by L. R. Ingersoll, O. J. Zobel and A. C. Ingersoll. (*McGraw-Hill*, 1948, p. 52).

is the line source. This may be done by equations differing from Equation 2 only in the value of the coefficient of the second term in the bracket and the substitution of a for r. The cases to which these equations apply are: (A), the true cylinder source (heat flowing to the cylinder surface from soil both inside and outside the cylinder, initially at uniform temperature); and (B), what may be called the isothermal core heat source (heat flowing to a cylinder from, or to, the outside only, the temperature inside the cylinder being kept uniform as the temperature of the cylinder is changed). In case (A) the resulting temperature of the cylinder surface may be calculated by

$$\theta = \frac{A}{5.4575\,k}\left[\log_{10}\frac{\alpha t}{a^2} + 0.21\,\frac{a^2}{\alpha t} + 0.351\right] \quad \ldots\ldots \quad (2a)$$

with an error of less than 1 percent for $\frac{\alpha t}{a^2} > 2$ (i.e., after 3 hours with a pipe of 3.82 in. diameter and soil characteristics as assumed).

For case (B), the so-called isothermal core case, the resulting temperature may be calculated by

$$\theta = \frac{A}{5.4575\,k}\left[\log_{10}\frac{\alpha t}{a^2} + 0.78\,\frac{a^2}{\alpha t} + 0.351\right] \quad \ldots\ldots \quad (2b)$$

with an error of less than 1 percent for $\frac{\alpha t}{a^2} > 6$ (i.e., after 9 hours with a pipe of 3.82 in. diameter and soil characteristics as assumed). It may be noted that Newman[14] developed the cylinder source formula (although he used it in examples which more properly fall in the isothermal core category). Carslaw and Jaeger[15] give the rigorous solution of the latter case. The above empirical forms bring out more clearly the fact that both cases converge and become identical with the line source for high $\frac{\alpha t}{a^2}$ values.

In the paper, certain temperature calculations were made using the line source formula Equation 2. It is to be noted, for example, that where times are in excess of about 2 days for pipes as large as 3.82 in. in diameter the results obtained with the line source formula differ negligibly from those obtained with the cylinder source[14] or isothermal core source[15] formulas. Hence, curves and conclusions in the paper are unaffected since times much longer than this form the basis of the present considerations. While these remarks might appear to suggest that the calculations from the line source consideration might have been dispensed with, it should be noted that the line source equation is needed for calculating soil temperature outside the pipe.

It is, of course, true as pointed out by Kayan that the idealized cases assumed for the purpose of calculations may fail to represent exactly any practical installation. It does make a difference that no heat receiving grid or plate can really be infinite in extent and the performance of a grid of a given area would certainly not be independent of the dimension of the grid. It is hoped, however, that the calculations for the idealized cases will provide reasonable starting points from which adjustments can be made on the basis of practical judgment.

Mr. Brown's remarks are valuable as calling attention to engineering considerations beyond those included in the paper which effect the prospect for the development of a generally available earth heat source.

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[14] Temperature Distribution in Internally Heated Cylinders, by A. B. Newman. (*Industrial and Engineering Chemistry*, Vol. 23, 1931, p. 19.)

[15] Heat Conduction, by L. R. Ingersoll, O. J. Zobel, and A. C. Ingersoll. (*McGraw-Hill*, 1948, p. 283.)

Heat Flow in the Region Bounded Internally by a Circular Cylinder, by J. C. Jaeger. (*Proceedings Royal Society of Edinburgh*, A, 61, 223, 1942.)

1369

No. 1369

HEAT PUMP RESULTS IN EQUITABLE BUILDING

By J. Donald Kroeker*, Ray C. Chewning† and Charles E. Graham**, Portland, Ore.

IN A PREVIOUS paper[1], it was stated *it is contemplated frank comparisons will be made between actual operating experience and costs as against those predicted from careful analysis during design.* Such comparisons are the purpose of this paper.

The Equitable Building, Portland, Ore., heat pump system, at present the largest in an office building in the United States, has been in operation for a complete year, although the building has not been fully completed or occupied during that period. Many requests for data on operation of this heat pump have been received. Complete data applicable to operation over a normal season coincident with normal building use are not yet available. However, sufficient data have been assembled to permit reliable comparisons which are not expected to change significantly as a result of further studies. Accurate cost comparisons cannot be made at this time for reasons which will be discussed.

The discussion of the operating results and comparisons is confined to those factors applicable to the heating phase, inasmuch as use of refrigeration equipment for the heating service is the only point of interest in the installation. Comments are also included on factors believed important in design of a commercial heat pump system.

Building and System Described

A brief general description of the building and system is necessary for this discussion. The building, Fig. 1, is a 12-story rental office building with a plan area 100 ft × 200 ft for the basement, first and second floors; 60 ft × 200 ft

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** Building Engineer, Equitable Building.

[1] A Heat Pump in an Office Building, by J. Donald Kroeker and Ray C. Chewning. (A.S.H.V.E. Transactions, Vol. 54, 1948, p. 221.)

Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Minneapolis, Minn., June, 1949.

for the third through the 12th floors; a 13th set back floor approximately one-half the latter area; and two penthouse levels housing air conditioning and elevator equipment. The building is of reinforced concrete construction, with aluminum facing. Double-pane windows extend from column to column. The

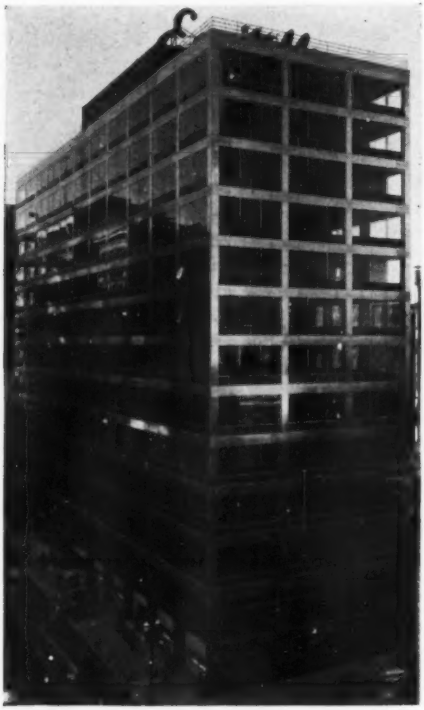

FIG. 1. THE EQUITABLE BUILDING, PORTLAND, ORE.

exterior pane is made of heat absorbing glass, which reduces the heat loss due to glass by 50 percent and the heat gain by 40 percent.

The entire building is completely air conditioned, using an all air system to avoid higher first cost of a split system. The heat pump which furnishes the heating and cooling to the air conditioning system is composed of four diffusion-type, centrifugal water-chilling machines, Fig. 2. Well water is the heat source. Wells not being pumped are used for disposal. The temperature control for air conditioning and the heat pump is somewhat more complicated than ordinary.

A flow diagram applicable to the heating phase below 55 F outside temperature is presented as Fig. 3.

SYSTEM COMPLETED BY STAGES

The building was placed in use progressively as tenants were secured and spaces completed for them (see Table 1). The heat pump was placed in

FIG. 2. MECHANICAL ROOM, WITH THE FOUR HEAT PUMPS AT RIGHT AND THE CONTROL AND INSTRUMENT PANEL AT LEFT

operation on January 9, 1948. For several months it was operated manually to supply the heat necessary for drying out plaster and to provide comfort for the few spaces which were at that time finished and in use in the building. By April 1948, automatic control for the heat pump had been placed in operation and recording of operating data was started. At that time the building was 48 percent occupied. There were still some areas of the building not closed to the outside. This was particularly true in the first floor area where shops and stores were later located. Also, in April of 1948 a preliminary adjustment of air volumes on the supply side only was accomplished. Final adjustments of air volumes were not completed until April 1949.

ACTUAL OPERATING METHODS

Operating methods varied in some respects from the usual. Following afternoon temperatures of 55 F or less, some heat pump capacity was provided throughout the night to reduce the morning pick-up load. Reducing pick-up load has two advantages in this installation. *First,* the heat pump, being less efficient at high condenser temperatures, could be operated at better efficiency

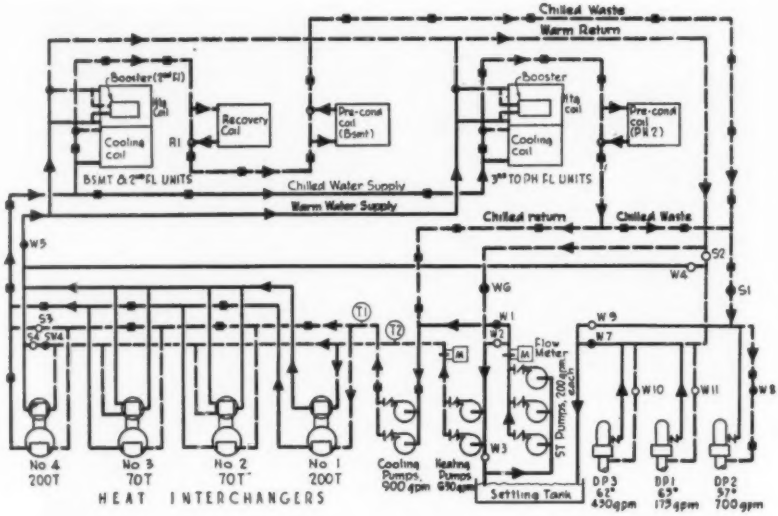

FIG. 3. FLOW AND CONTROL DIAGRAM

(When heating is greater than cooling load)

during the pick-up period. *Secondly,* the pick-up period was reduced, thereby avoiding creation of an electrical demand which would be superimposed on the normal building demand and result in a demand charge under the electric tariff applicable. The method of operation during cold weather was as follows:

During the night, starting at about 10:00 p.m. until about 6:00 a.m., either a single 70-ton machine, or, if the outside temperature was below 30 F, two 70-ton machines were placed on manual operation and allowed to run through the night. All outside air intake fans and exhaust fans, except for the toilet room group, were turned off. At approximately 6:00 a.m. all machines were placed on automatic operation and outside air again supplied. This continued until approximately 6:00 p.m., at which time the outside air supply was turned off. The machines remained on automatic operation until 10:00 p.m., at which time the cycle was repeated.

The amount of well water used as a heat source was considerably less than anticipated, and conditions during which exclusive interchange of heating and

cooling prevailed extended over a greater period than expected. On days during which the average temperature was between 44 and 57 F and the diurnal range was relatively low (a condition typical of cloudy weather operation) well water was required by the heat pump system and automatically injected into it only during the morning pick-up period. Beyond this time period the heat losses and heat gains were sufficiently balanced to require no additional well water or, at most, very little well water. Since Portland's average winter temperature is 49.5 F and cloudiness is prevalent during the temperature range previously

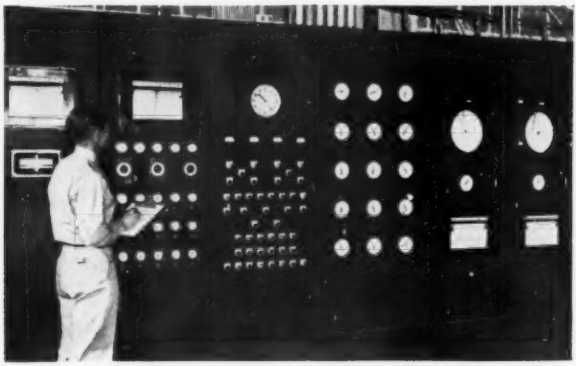

FIG. 4. INSTRUMENT AND CONTROL PANEL

Showing room temperature recorders at left, pneumatic room thermostat adjusting switches and fan controls (in front of operator); main heat pump controls below clock; pressure and temperature gages; and flow meters

indicated, the days on which little or no well water was used account for a very material fraction of the heating season.

INSTRUMENTATION APPLIED

Detailed data on operation of the heat pump were obtained for the most part by recording instruments, Fig. 4. Pertinent data not recorded were read and entered on a daily log sheet.

Each of the three water circuits (heating, cooling, and well water) introduced into the system was provided with a recording-integrating flow meter, Fig. 5. Air temperature recorders, accurate to one-half degree, were provided for outside air and five space temperatures for each of three building floors. Water temperatures were recorded (1) at the combined supply to and discharge from condensers and evaporators; (2) at the discharge of each deep well; and (3) in well water introduced into the system, Fig. 5. Water temperature recorders

were accurate to one-fifth degree. Indicating thermometers were used to check and verify recorded temperatures. From recorded data obtained on these instruments, heat output from the system and heat released within the building were computed.

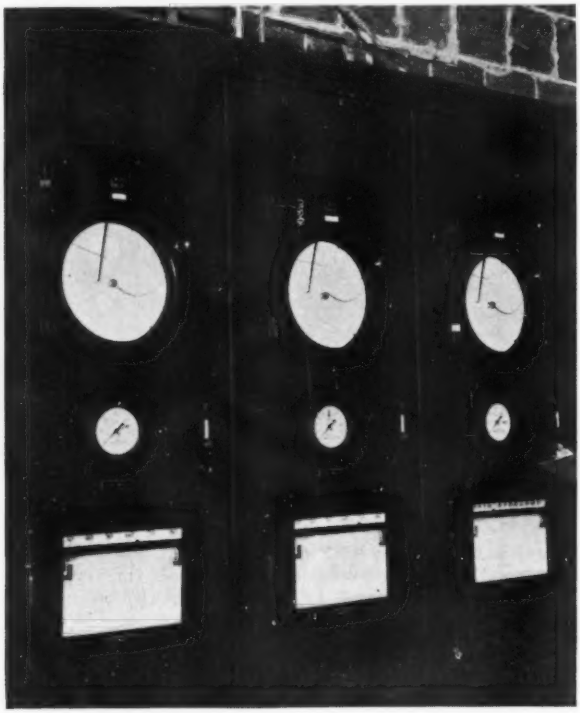

FIG. 5. FLOW METERS AND WATER TEMPERATURE RECORDERS

Flow meters give record of water flows from settling tank and in heating and cooling circuits. Multipoint recorders provide data on water temperature

A temporary panel, Fig. 6, contained electrical input and demand recorders for the water chilling units, two for deep well pumps, settling tank pumps, warm water circulating pumps, and chilled water circulating units. By means of these recorders, data were obtained to determine power input for the components of the heat pump system.

Conditions for which temperature data were not recorded continuously include: water entering the recovery coils; water entering the pre-conditioning

TABLE 1. PREDICTED AND ACTUAL POWER USE

(TOTAL FOR COMPRESSORS AND ALL PUMPS, IN 100 KWHR)

MONTH	PREDICTED[a] 1940	PREDICTED[b], ADJUSTED TO 1948-49	ACTUAL 1948-49	BUILDING OCCUPANCY (PERCENT)
May	830	725	624	56
June	1033	1110	1080	62
July	1065	1060	993	74
Aug.	1253	1135	1080	89
Sept.	909	932	930	91
Oct.	705	730	809	92
Nov.	720	854	809	93
Dec.	720	1112	894	94
Jan.	814	1650	1640	95
Feb.	689	1098	1045	95
March	627	812	705	95
TOTALS	9365	11,218	10,609	

[a] Predicted use based upon Portland Weather Bureau Charts for 1940.

[b] Predicted values corrected for degree days and hours of operation, for 1948-49.

coils; and water leaving the pre-conditioning coils. However, with air volumes and temperatures known and with data on water flow recorded, sufficient information was available to determine within reasonable limits the amount of heating accomplished by means of waste chilled water.

BASIC CONDITIONS COMPARED

Predicted power consumption and power costs were based on Weather Bureau data for the City of Portland for the calendar year 1940. This year was selected, because, during design, a consideration of the necessity for dehumidifi-

TABLE 2. ACTUAL TOTAL HEAT INPUT AND OUTPUT FOR SELECTED DAYS

(24-hr average Mbh)

ITEM	DATE	AVERAGE TEMPERATURE	CONDENSER HEAT OUTPUT	HEAT INPUT	
				COMPRESSORS ONLY	COMPRESSORS & AUXILIARIES[a]
1	Jan. 20	28	2240	566	623
2	Jan. 25	22	2510	623	715
3	Jan. 26	23	2500	623	702
4	Jan. 30	36	2000	453	510
5	Feb. 2	34	2220	510	554
6	Feb. 8	40	2030	398	418
7	Feb. 16	51	1570	283	306
8	Feb. 17	44	1658	398	442
9	Feb. 18	36	1930	453	500
10	Feb. 21	41	2020	458	479

[a] Deep well pumps and settling tank pumps.

cation required a detailed study in which hourly weather conditions, including dew point and dry bulb temperatures, were plotted for six months of the summer season. These studies showed 1940 as a critical year and provided much detailed data useful in cost analyses. The heating degree-days for 1940 were below normal. The winter of 1948-49, for which the operating data are avail-

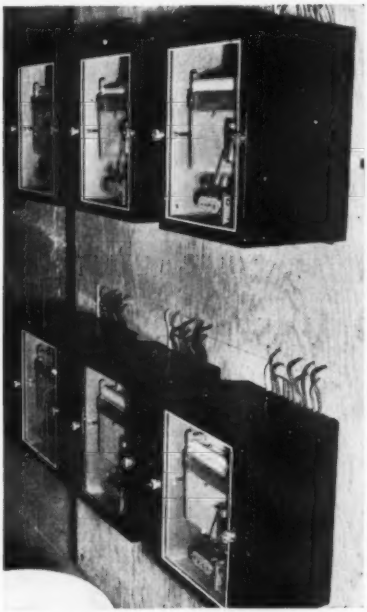

FIG. 6. TEMPORARY WATT-HOUR AND DEMAND METERS

Record input data to heat pumps and collateral equipment

able, was probably one of the most rigorous Oregon has experienced. While 1940 had 4100 degree-days, 4900 degree-days are reported for the 1948-49 winter and comparisons herein are based on these values.

The outside temperatures used in this study were Portland average daily temperatures. The days selected for study and computation of heat balances were days on which there was no sunshine. The loads shown are, therefore, somewhat higher than normal, since Portland's winter days, though predominantly cloudy, do have considerable sunshine.

The 24-hour average temperature method for determining heating requirements was considered proper because of the building lag involved in transmission

of heat. An example is the transmission of heat from lights and other electrical input involved to the building materials, such as the ceiling or concrete floor slab, before it is transmitted to the air or affects the occupants of the building. Use of the 24-hour average temperature method takes into account and provides

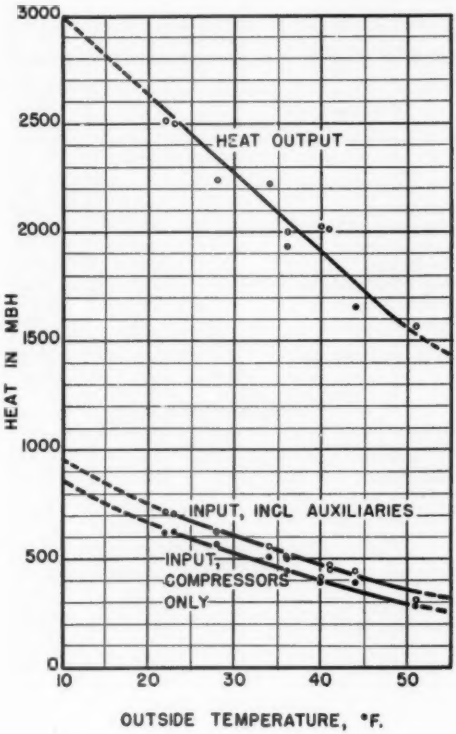

FIG. 7. HEAT BALANCE CURVES

some allowance for the fact that steady state conditions are practically non-existent.

POWER REQUIREMENTS COMPARED

A comparison of predicted and actual power requirements for the compressors or water chilling units and all pumps for eleven months of operation is shown in Table 1. To provide a proper basis of comparison, the predicted values, based on 1940 weather, have been adjusted to 1948-49 conditions and for differ-

ences in operating methods, as described. The latter was estimated as equivalent to three hours of daily additional operation for the colder weather. Power requirements were 5.3 percent less than adjusted predicted values. No adjustment has been made in the values for hours of operation in the months of May to October.

TABLE 3. HEAT BALANCE FOR SELECTED DAYS

(24-hr average Mbh at average outside temperature)

ITEM	CONSIDERATION	FEB. 16 51 F	FEB. 21 41 F	FEB. 2 34 F
	Heat Released in Building			
1	Well Water	180	540	690
2	Compressor Motor	283	396	510
3	Pump Motors	80	100	140
4	Fan Motors	150	150	175
5	Bldg. Electrical Use	1020	1020	1020
6	Occupants	245	245	245
7	Recovery	25	275	300
8	Total Heat Released	1983	2726	3080
	Heat Supplied by System			
9	Pre-Conditioning Coils	125	350	475
10	Heat Pump	1570	2020	2220
11	Total Heat Supplied	1695	2370	2695
	Heat Balance			
12	Heat Released	1983	2726	3080
13	Heat Supplied	1695	2370	2695
14	Vagrant Loss	288	356	385

HEAT BALANCES ESTABLISHED

The actual total energy input and output were computed for selected days from charts recorded as described, Table 2. Days selected for study were chosen from those having complete and unbroken records. The data are shown graphically in the heat balance curves of Fig. 7 and form the basis for establishing a general graphical comparison of heating requirements and loads. Insufficient study of all records and data was made to establish fully the heat input and output curves of Fig. 7. Some data for days having lower average temperatures were faulty, precluding more complete analyses. The heat input lines are shown as curves chiefly on the basis of lowered thermal efficiencies at the higher supply or condenser temperatures required at lower outside temperatures.

To arrive at a full heat balance, examination was made of all sources of heat in the building, including well water and motor equipment. Typical tabulations for three days are shown in Table 3. Heat gain from occupants was taken at 33 percent of the sensible heat of actual occupants for business hours, inasmuch

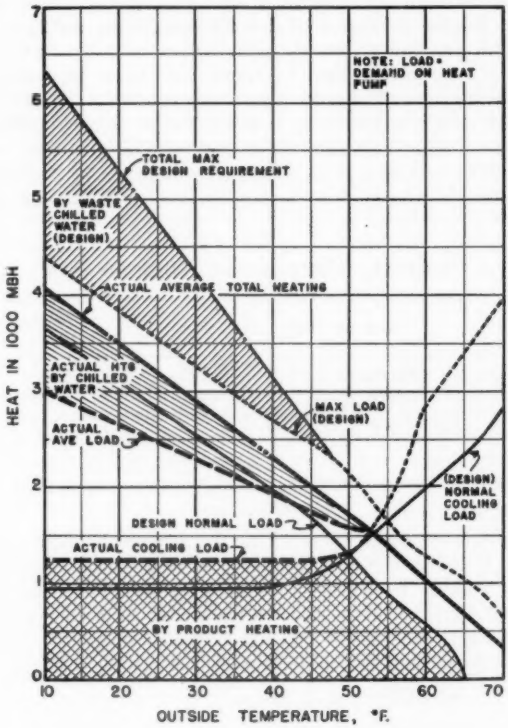

FIG. 8. COMPARISON OF PREDICTED AND ACTUAL HEAT REQUIREMENTS AND LOAD

as the consideration is on a 24-hour basis to establish average values. Other heat gain values were obtained from measurements. Vagrant or unaccounted-for losses range between 12.5 and 14.5 percent of values for total heat released in the building at the temperatures given.

DETAILED COMPARISON MADE

The comparison of heating requirements and loads on the heat pumps predicted during design and actual conditions, as determined from the data and by

means indicated, is shown in Fig. 8. Except that values for cooling are shown only as they affect heating, Fig. 8 consists of load curves presented in Fig. 1 of the previous paper[1], upon which are superimposed data on actual operating characteristics.

An important difference between predicted and actual heating curves shown is that the predicted curves were based on instantaneous values and the actual were based on 24-hour average conditions of temperature and load, as previously discussed. This accounts at least in some measure for the fact that the curve, *actual average load,* shows values of appreciably lower magnitudes. Similarly the amount of heat gained by recovery from exhaust ventilation air and use of waste chilled water to pre-heat or pre-condition incoming ventilation air is naturally of lower magnitude when considered on a 24-hour average basis.

The slope of the *actual average load* curve is less steep than that predicted. This may be due to mixing of air in the double plenum systems used primarily

TABLE 4. COEFFICIENTS OF PERFORMANCE

ITEM	CONSIDERATION	COMPRESSORS ONLY	COMPRESSORS & AUXILIARIES[a]
1	As operated, with credit for by-product heat....	22.6	19.2
2	As operated, without credit for by-product heat.	5.6	4.8
3	Without credit for heat recovery and heating with waste chilled water.....	4.9	4.2

[a] Deep well pumps and settling tank pumps.

for control purposes, particularly when the outside temperature was in the vicinity of 50 F, when cooling and heating loads are approximately in balance.

The actual cooling load curve, representing by-product heat available, is higher than predicted. It is based on a study of the amount of electrical energy used throughout the building in applications other than air conditioning, as detailed for three conditions in Table 3.

COEFFICIENTS OF PERFORMANCE REVISED

The previous paper[1] indicated a job coefficient of performance of 14.8 as fantastic under usual interpretation placed on the term. Under study of day-by-day average temperatures from May 1948 through March 1949, the integrated load between the area shown as *by-product heating* and the *actual average load* curve plus the heating accomplished with waste chilled water, when related to heat input, develops the even more fantastic coefficient of performance of 22.6 for compressors only, Table 4. Coefficients of performance on other bases are also shown in this tabulation.

The heat input for auxuliaries includes only the heat equivalent of power used by the deep well pumps and the settling tank pumps on the basis that a heat pump merely replaces the conventional heating means, such as a boiler, and thus power required by equipment inherent in the remainder of the system,

or the heat distributing means, is not chargeable against the heat pump. Thus, the warm water and chilled water circulating pumps and the fan equipment are not considered auxiliaries in this paper.

The coefficients of performance presented in Table 4 are another indication, previously suggested, that in any complex heating and cooling system a coefficient of performance is without meaning in the absence of detailed specification of division of load and means of application.

The justification for taking credit for by-product heating is the arrangement of the air conditioning system.

Outside air was not used for the internal or minimum cooling, since to do so would have required equipment arrangement on each floor, reducing the rentable space materially, and would have required even more complex automatic control. Thus mechanical cooling provides by-product heat and is considered properly credited to the heat pump. If use of steam for heating had been planned, with the heat pump adapted separately for cooling, the internal cooling, or the minimum cooling requirement shown in Fig. 8, would have been accomplished with outside air, reheated as required.

COSTS NOT ANALYZED

As discussed in the previous paper[1], the electric utility's rate schedule applicable provides charges on both power consumption, and monthly maximum 30-min sustained electrical demands. Power consumption for each of the heat pump motorized elements and the total for the building were recorded, as were the demands for the heat pump units. However, the type of recording demand meter required for the building's electrical service was not available until quite recently. The weather has been mild since installation of the recording demand meter, so there has been no opportunity for records indicating whether the heat pump operation superimposes a demand which will be higher than the maximum building demand and thus be chargeable to the heat pump. Sufficient records are available to show that for outside temperatures ranging from 40 F upward, a uniform demand, higher than the building demand is created. Electrical demand of the heat pump for morning pick-up occurs at a time prior to building occupancy and is much smaller than normal building demand. It is quite possible, however, that heat pump electrical demands during colder weather will superimpose on building demands and create peaks higher than those of record. Since such demand is subject to a charge of $1.00 per kilowatt, the charge against heating may be material. Because such probable additional demand due to heat pump operation is not subject to accurate estimation, and since it may materially affect operation cost, although certainly not of the magnitude anticipated by the previous paper[1], no reliable data can be reported on heating cost at this time. However, from indications it is quite certain that the heating cost will be materially less than the $0.46 per million Btu originally estimated.

GENERAL OBSERVATIONS

Observation of a year's operation gives rise to comments which may be of interest and of some value in design or operation of a heat pump in a large building.

The heat source normally adapted will probably be well or surface water. Often such water is corrosive or may form accretive substances on metals. Water analyses should be made in all cases to determine required treatment, if any. For example, two wells in the Equitable Building have water which is slightly acid and which contains appreciable amounts of iron and of carbon dioxide. Corrosion has resulted and iron bacteria have caused some problems. Both are now under simple but adequate treatment. Necessity of treatment should be determined prior to application of design, since in some cases the cost thereof may be so high as to render application of a heat pump uneconomical.

One of the primary causes of corrosion from well water is the oxygen contained in the water as it is introduced into the system. Even with water which otherwise might be considered good water, corrosion will result if the water is aerated to any degree. Therefore, provision should be made to limit to a minimum the amount of air picked up by the water. This requires a type of settling tank which does not present a large surface of water exposed to air. It also requires constant pumping of the well, with modulation of volume obtained by the use of valves.

The feasibility of heat recovery with the heat pump systems due to the quantities of water and air being handled is definitely substantiated by experience of operation on the Equitable Building, as is the economy of using the waste water for pre-conditioning incoming ventilation air.

The importance of applying the best possible automatic control in the interest of operating economy cannot be over-emphasized. Because it is necessary to use small temperature differentials in the supply and return water and in the supply air, the control must be accurate and it must be flexible. Flexibility is required in order to maintain the most efficient conditions of operation for the refrigeration equipment. For example, while a coefficient of performance of 5 or 6 may be obtained on a basic refrigeration cycle with a condensing temperature of about 80 to 85 F, the coefficient may drop to as low as 3 when this temperature is raised to 105 or 110 F. The major capacity control instruments in any heat pump installation of over 200 tons capacity should be of the industrial type, with automatic reset within the instrument, if the method of control requires it.

In providing for the automatic change of water circuits from one type of operation to another, care must be exercised in arranging the opening and closing of valves in sequence to prevent damage from water shock or hammer.

Control devices should be located in a central mechanical room with the primary heat pump equipment, and sufficient recording instruments of various types should be installed. Such instruments include flow meters, temperature recorders, room temperature and humidity recorders, together with units to control the outside air fans, exhaust fans, etc. With assistance of this nature, an intelligent operator can hold the air conditioning costs to a minimum, observe operation of his equipment, and know the comfort conditions in the building with very little additional manual help.

The operator himself should be above the average operating personnel. He must have an analytical mind which can sort out and properly correlate the information which the recorders give. An intelligent operator properly trained

can save $150 to $200 a month in operating costs alone in a system the size of that in the Equitable Building.

Finally, observation of the Equitable Building heat pump application has indicated that the carefully detailed studies of loads and methods, including several innovations, have paid dividends in economy of operation.

DISCUSSION

E. W. GUERNSEY[2], Baltimore, Md. (WRITTEN): This is a very excellent report. Dealing, as it does, with details of design and experience with a very large heat pump system, it should provide encouragement to others to consider the heat pump possibilities in other particular situations.

I am wondering in how many places, a well water supply would be available in quantities such as used in this system and at temperatures over 60 F. Can anything be said as to the geology in the region of the building involved which would indicate whether or not there are exceptional factors which would not be generally present? It is interesting that the wells (as shown in an earlier paper[3]) at 150 ft are 5 to 8 deg warmer than the one at 510 ft. Is there a known reason for this?

I am not certain that my understanding of the significance of the figures in the first line in Table 4 is correct. I assume that, for example, 22.6 is the ratio of the total heat supplied, to the excess of electric energy to the compressor over that which would be required for cooling only, and which is therefore chargeable to heating. This is perhaps a useful ratio. Some would, however, hesitate to call it a *coefficient of performance*, a term which has commonly been used as an index of the effectiveness of a given motor-compressor combination in moving heat from a lower to a higher level.

G. E. MAY, New Orleans, La. (WRITTEN): The electric utility has a natural interest in the heat pump because it converts the average compression type air conditioning system from a poor load factor problem to a good load factor and profitable load. Many utility engineers have followed the development of the heat pump with great interest and have contributed heavily toward early research efforts.

We, in New Orleans, thought that the warm climate in this region, along with the many sources of high temperature water during the winter months would make the heat pump very practical and result in performance ratios higher than those in northern sections. Here, it would be a *natural*. But, economics play an important role in the selection of year-round air conditioning equipment. We found that, throughout many parts of the Middle South area and large adjoining regions, the heat pump suffered in both investment and operating cost comparison with gas operated year-round systems. This is due to the fact that the Middle South is rich in natural gas resources, where an abundance of 1100 Btu gas is available at a very low cost. Even where the latest steam generating techniques are employed, electricity cannot compete with gas for such heating services.

Another equally interesting observation in this Middle South area is the performance failure of the air to air model heat pump units. Even in small commercial applications, where some internal sensible heat from illumination was available, the heat pump failed to perform adequately when the outside temperature dropped below 45 F. Supplementary heat was required. Tests in residential work indicated a greater shortage of capacity for winter heating.

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[2] Director of Research, Consolidated Gas Electric Light and Power Co. of Baltimore.

[3] A Heat Pump in an Office Building, by J. Donald Kroeker and Ray C. Chewning. (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 221.)

On the other hand, the water to air units seem to work very satisfactorily. Deep wells (800 ft strata) producing 68 to 72 F water, provide the best heat source.

E. P. PALMATIER, Syracuse, N. Y. (WRITTEN): I am sure those of you who have followed the previously published material on the heating and cooling system for the Equitable Building in Portland will agree with me that this paper is a welcome progress report on the operating results obtained. It is not particularly significant that, while the title of the paper implies that the system installed in the Equitable Building is a heat pump system, in fact it is not a simple heat pump system at all. Neither is it significant that the system is not a simple cooling or air conditioning system. The significant fact is that *in the Equitable Building we have a fine example of the application of good engineering to the overall heat balance of an office building wherein the greatest economy was achieved by eliminating the use of fuel burning equipment for heating.*

The authors refer to the refrigeration machines as *diffusion-type centrifugal water-chilling machines.* I would like to inquire into the origin of this terminology and the reason behind it.

I would like to have the author clarify Fig. 7. The figure is labelled *Heat Balance Curves.* The text does not make clear what sort of heat balance is involved. If I were to guess, I would say that the upper line in Fig. 7 is the heat output of the condensers of the refrigeration machine plus perhaps the energy equivalent to the power put into the warm water pump. The two lower curves are clearly labelled. Since the ratio of corresponding values taken from the upper curve and two lower curves would be the heating coefficient of performance of the refrigeration equipment either with or without the energy input to auxiliaries, I feel that the figure should have a title in which the words *heating COP* are included.

Table 3, titled *Heat Balance for Selected Days,* is not at all clear. If this table is supposed to represent a heat balance on a building, where is the item for heat loss (or gain) from outside the building either by transmission or otherwise? Since there is no such item we might conclude that the 24-hour average heat transfer between the building and outside was essentially zero in each case. If there were no net heat exchange between the building and outside, then it is clear how items 1 to 8 in the upper part of the table would nearly equal the heat release from the condensers of the refrigeration machines, which is represented presumably by item 10. I would expect, however, that heat gains of the building, particularly solar heat gains, could be very sizable items and if they were not balanced almost exactly by transmission heat losses any attempt at making a balance such as is made in Table 3 would be met by failure. Would the authors please clarify the meaning of Table 3.

It would appear from the picture of the building, Fig. 1, that the exterior walls are almost 50 percent glass. Considering this fact, I am surprised at the statement of the authors that the *24-hour average temperature method of determining heating requirements was considered in order because of the building lag involved in transmission of heat.*

Finally, in reading the paper I was very confused by Fig. 8. Some of the confusion arises from the labelling. Specifically, we find the label *Actual Heating by Chilled Water.* The upper shaded area is labelled heating *By Waste Chilled Water (Design).* The lower shaded area labelled *By-Product Heating* is bounded at the top by a dash curve labelled *Actual Cooling Load.* Would the authors please clarify whether the by-product heating which is provided by the cooling load and which is apparently constant below approximately 45 F outside temperature includes the power input to the refrigeration equipment corresponding to the cooling load. In other words is the value of 1.25 shown in terms of condenser heat or evaporator heat? If it is condenser heat, it would seem impossible to have the curve continuous with the cooling load curve to the right of 50 F outside temperature. If it is evaporator

heat, it would seem that the authors have not taken credit for as much by-product heating as they might possibly take credit for.

In closing I want to say that I hope the authors will not feel that my remarks are intended to be critical of their work. I can well imagine the difficulty of presenting a clear description of the design and operation of such a complex system as that installed in the Equitable Building. I was honestly puzzled by the things which I have mentioned and I would appreciate very much if these few points might be clarified.

AUTHORS' CLOSURE: With reference to Mr. Palmatier's comments, the term *diffusion-type centrifugal water-chilling machines* was used to describe a type of equipment containing a centrifugal compressor discharging gas from its impeller to a chamber as against one which discharges to a volute scroll.

The labelling on Fig. 7 could have been more specific. The heat output curve actually shows the heat output of the heat pump. The figure as a whole shows heat balances of the compressors only and of the compressor with auxiliaries. It would be incorrect to caption the figure *Heating COP* because the curves shown are not COPs, but do contain some of the data from which COPs have been calculated. An additional curve might be added to the figure to give the complete data from which COPs could be calculated. This curve would be titled, *Total Heat Furnished to the Building by the System,* and would be plotted above the heat output curve by an amount equal to the heating accomplished by recovery and waste-chilled water.

Table 3 is not intended as a heat balance showing the heat required by the building and the heat supplied to the building according to normal concepts of heating. It is a heat balance for the total of all sources of heat available in the building and the amount of that heat utilized in the heat pump. A large share of the so-called vagrant losses are actually used in the exterior zone to offset the heat losses directly. To the design engineer, the most significant fact learned from the first year's operation of the Equitable Building is the quantity of the various internal heat sources in the building. This fact is important for two reasons; namely, *first,* it shows the increasing need of year-'round air conditioning for this type of building, which in turn makes the heat pump a desirable system if power rates and heat source factors are favorable, and *secondly,* it greatly reduces the outside heat source requirement, which is one of the major problems in heat pump application. The solar heat mentioned in the discussion is not a factor because, during heating weather, it merely acts to reduce quantity of heat distributed to the exterior of the building. Fig. 8 indicates the comparison between calculated heat loads and actual heat loads. Since it is rather difficult to place a building the size of the Equitable Building in a device giving results expected from a calorimeter to determine its gain and loss of heat due to outside conditions, and since this was a presentation of actual measured data, no attempt was made to recalculate heating loads.

The reasons for using the 24-hour period instead of hourly periods are apparent in consideration of the heat gain from lights, which transmits to the building materials and is later transmitted as a cooling load. Reference is made in this connection to the paper[4] by Professors Mackey and Gay, also presented at the Minneapolis Meeting.

The area labelled *by-product heating* in Fig. 8 is based upon the cooling load present in the interior of the building. It does not include the compressor motor heat required to carry that load. The area was plotted as shown to simplify as far as possible this rather complicated chart. Since this procedure results in a lower calculated job COP, it was thought to be permissible to apply this simplification for clarity.

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[4] Heat Gains Are Not Cooling Loads, by C. O. Mackey and N. R. Gay. (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 413.)

I was gratified to see someone make as thorough a study of this paper as Mr. Palmatier obviously did. The fact that he did, and that others may be caused by our paper to hang on to British thermal units once they get them in a building, will more than repay us for the long hours required in coordinating the recorded data into a form at all presentable.

An odd thing it is, but true, that engineers will find resistance to conservation of heat normally wasted in applying cooling to buildings. Just recently, in designing a cooling system requiring 250 tons capacity in the first stage of construction, we were not allowed to similarly harness the roving Btu's because to do so meant it would be a heat pump job, and in our client's conservatism the heat pump was considered a device too new to hazard an application. Probably we should have taken the cue from the first paragraph of Mr. Palmatier's discussion and called it something else since, as you may have observed, wasting heat that can be economically recovered causes us to wince

1370

No. 1370

GROUND TEMPERATURES AS AFFECTED BY WEATHER CONDITIONS

By A. B. ALGREN*, MINNEAPOLIS, MINN.

GROUND temperature data on the various soils for different climatic conditions are becoming increasingly important in many phases of heating, ventilating and air conditioning engineering. Some of the more recent developments for which these data might have direct application would be (*a*) the earth as a heat source for the heat pump application; (*b*) the perimeter and ground slab loss for floor panel heating systems; and (*c*) the ground losses together with floor slab temperatures for basementless houses where other heating methods exclusive of floor panel systems are used.

The tests reported in this paper were part of a two-year research project covering a field study on floor panel heating systems. The ground temperature study was made on a 110 acre plot of ground which is located at Edina, Minn., and which was selected as the site for a large residential building project.

In addition to the ground temperature data, laboratory and field tests were made on the soil to determine its mechanical analysis, percolation or permeability value, moisture content, density, specific heat and conductivity. Field tests were made on the soil for determining its density and percolation value. The mechanical analysis, moisture content, specific heat and conductivity were determined by laboratory tests on an average sample of the soil collected at depths of from 1 to 3 ft. In order that representative results might be obtained, approximately 200 lb of soil were examined.

During the first year, temperatures were determined by thermocouples inserted in test holes drilled to a depth of 8 ft. It was evident from this study that if information of practical value on ground temperatures was to be obtained, it would be necessary to take temperatures at depths considerably below the 8-ft level. At the 8-ft depth the temperature variation was approximately 21 deg from August to May. Therefore, during the 1947-1948 year, new test holes were drilled and temperatures were recorded to a 16-ft depth.

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Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Minneapolis, Minn., June, 1949.

TEST HOLES DRILLED

Ten test holes were drilled with the purpose of finding the locations in which the soil was approximately of uniform consistency throughout the depth tested. When the first several test holes were drilled, a stratum consisting of a mixture of silt, clay and about 60 percent free water was encountered at the 7-ft depths. This stratum was about 9 in. thick and was found to exist in a low lying area in the vicinity of a small pond. As all of the sod and brush had

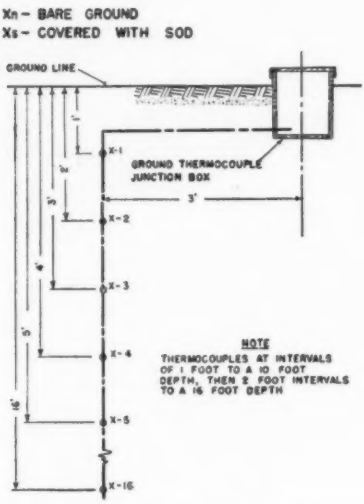


FIG. 1. THERMOCOUPLE LOCATIONS (SERIES X_n, X_s)

been cleared away from this area, it was decided to locate one series of thermocouples at this point.

This location provided a means for studying the effect of barren ground and also what might be called ground surface water. The thermocouple series referred to as X_n in this paper was located at this point. Other test holes showed uniform soil consistency throughout their depth. As part of this area was covered with brush and a thick, matted sod, this site was selected for the second series of thermocouples referred to as series X_s. The two thermocouple series were approximately 1200 ft apart. These two conditions provided a means for studying the insulating effect of sod.

A 1½-in. diameter soil sampling auger was used for drilling the test holes.

The thermocouples consisted of 24 gage, copper-constantan wires with a soldered junction. The thermocouple wire was double cotton covered, enameled wire.

In order to avoid any short circuiting of the thermocouple wires due to ground moisture, the investigators encased each thermocouple in a small diameter, moisture and vapor resistant rubber tube. The end of the rubber tube at the thermocouple junction was heat sealed; each thermocouple was similarly pro-

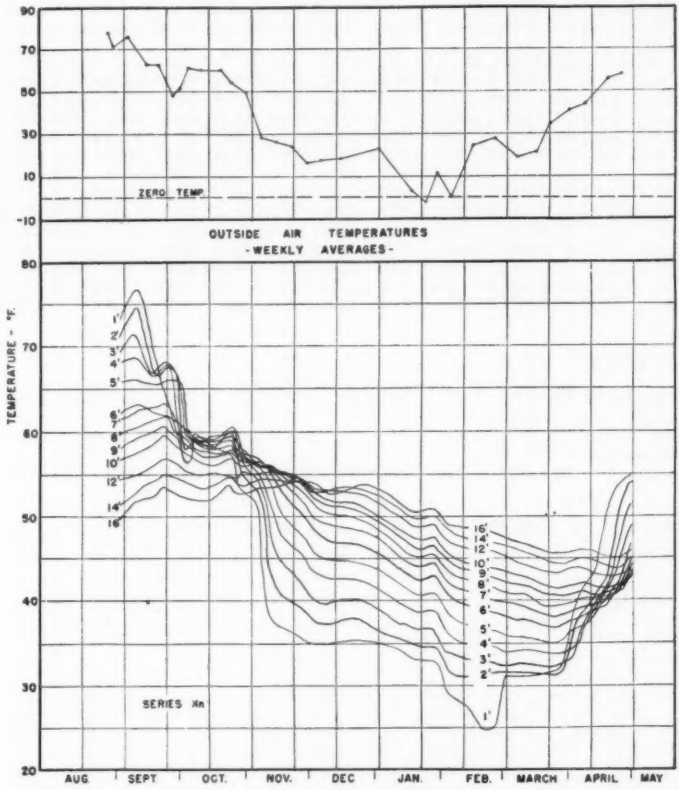


FIG. 2. GROUND TEMPERATURES AT DIFFERENT DEPTHS (AUGUST 1947 TO MAY 1948)

tected from its junction to the thermocouple junction box. Preliminary tests indicated that this procedure gave adequate protection.

Each thermocouple series was completely assembled before being placed in the test holes. The thermocouple junctions were accurately spaced at measured intervals and taped together as a single unit. The end of the thermocouple assembly that was to be located at the 16-ft depth was then fastened to a slip-

TABLE 1. GROUND TEMPERATURES

SERIES X_n—BARE GROUND IN SERIES

DEPTH BELOW SURFACE (Ft)	AUG. 1947	SEPT. 1947												
	21	2		9		18		23		30	AVERAGE			
	X_n	X_s	X_n	X_s	X_n	X_s	X_n	X_s	X_n	X_s	X_s	X_n	X_s	X_n
1	75.0	67.3	73.2	68.2	76.6	70.1	68.9	64.4	64.5	58.2	55.5	70.8	63.2	56.5
2	74.7	62.8	71.0	64.3	74.5	67.0	66.6	62.5	67.6	59.7	58.4	69.9	61.6	56.4
3	73.1	60.1	69.0	62.0	71.3	64.0	66.6	61.4	68.1	59.3	58.9	68.8	60.4	57.9
4	71.2	57.2	67.9	59.6	68.4	61.5	66.5	60.4	67.8	58.5	59.5	68.1	59.2	59.5
5	68.3	55.3	65.8	57.3	66.0	60.0	65.5	59.1	66.0	56.8	59.3	65.8	57.8	60.2
6	64.3	53.9	62.2	55.7	63.1	58.5	62.2	58.8	61.9	55.5	59.1	62.3	56.9	60.2
7	61.5	52.3	61.7	54.4	62.5	56.5	62.8	54.9	63.2	54.2	57.1	62.5	55.0	59.9
8	59.0	51.2	59.9	53.4	60.5	56.0	61.4	55.1	61.9	53.3	57.9	60.4	54.5	59.2
9	57.0	50.1	58.2	52.3	59.3	54.6	60.1	54.1	60.7	52.2	56.5	59.6	53.5	58.5
10	54.9	49.2	56.9	51.0	57.5	53.5	58.6	53.6	59.6	51.3	56.2	58.2	52.6	57.5
12	52.0	48.0	54.5	49.3	55.0	51.5	56.2	50.2	56.9	49.5	54.0	55.6	50.4	55.5
14	49.8	47.0	51.4	48.0	53.0	50.0	54.1	48.6	55.0	48.2	53.1	53.4	49.1	53.9
16	48.0	46.3	49.9	47.5	51.9	49.2	52.4	48.1	53.6	47.1	52.2	51.9	48.3	52.2
Depth of snow on ground in.	...	...	...	...	...	...	...	...	...	...	...	...	...	...
Mean Air Temp	85	85	74	74	84	84	78	78	60	60	42	63.1	63.1	66
Avg Air Temp for Preceding Week	78.1	78.1	71.2	71.2	76.4	76.4	63.4	63.4	62.6	62.6	48.4	...	...	51.3
Percent Sunshine	...	100	...	98	...	93	...	51	...	95	40	...	64	...

DEPTH BELOW SURFACE (Ft)	NOV. 1947										
	4		11		18		25		AVERAGE		
	X_n	X_s	X_n	X_s	X_n	X_s	X_n	X_s	X_n	X_s	X_n
1	52.1	51.2	38.4	42.3	36.8	39.8	36.0	39.5	40.8	43.2	34.8
2	53.6	52.0	44.2	46.6	40.6	42.9	39.6	42.2	44.5	45.9	37.6
3	55.0	52.7	48.9	49.7	44.3	45.5	42.8	44.9	47.7	48.2	40.2
4	55.8	53.1	51.9	51.9	47.7	48.0	46.0	47.3	50.3	50.1	43.3
5	56.7	53.2	54.3	52.7	50.3	49.6	48.7	49.0	52.5	51.1	45.7
6	57.2	53.2	55.5	53.3	52.9	50.6	51.0	50.3	54.2	51.8	48.3
7	57.2	53.1	56.0	53.4	54.1	51.4	52.7	51.1	55.0	52.3	49.8
8	57.0	53.0	56.1	53.2	54.9	51.7	53.7	51.8	55.4	52.4	51.3
9	56.6	52.8	56.1	52.8	55.1	51.8	54.4	52.3	55.5	52.4	52.2
10	56.4	52.5	56.0	52.8	55.2	51.8	54.7	52.5	55.6	52.4	52.8
12	55.3	51.8	55.2	52.2	55.0	51.4	54.8	52.3	55.1	51.9	53.3
14	54.1	51.0	54.5	51.7	54.4	50.8	54.5	52.1	54.4	51.4	53.0
16	53.2	50.5	53.4	51.0	53.7	50.4	54.0	51.9	53.6	50.9	53.0
Depth of Snow on Ground in.	....	....	1.8	1.8	3.5	3.5	3.7	3.7	....	....	3.5
Mean Air Temp	50	50	22	22	30	30	16	16	26.0	26.0	27
Avg Air Temp for Preceding Week	50.0	50.0	28.4	28.4	26.3	26.3	24.1	24.1	....	....	16.7
Percent Sunshine	....	0	....	24	....	0	....	58	....	19	....

—AUGUST 1947 TO APRIL 1948

X_s—GROUND COVERED WITH SOD

OCT. 1947											
3		7		14		24		28		AVERAGE	
X_s	X_n	X_s	X_n	X_s	X_n	X_s	X_n	X_s	X_n	X_s	X_n
56.5	55.0	59.7	56.1	58.4	54.9	57.4	54.2	53.5	52.0	56.9	54.3
56.4	54.8	59.1	55.6	58.2	54.5	60.0	54.9	56.0	53.7	58.3	54.7
57.9	55.6	59.1	55.4	59.0	54.3	60.5	54.9	57.4	54.0	59.0	54.7
59.5	56.1	58.8	55.1	59.3	54.5	60.1	54.8	58.3	54.0	59.1	54.6
60.2	56.1	58.8	55.0	58.8	54.5	59.7	54.8	58.3	54.0	58.6	54.6
60.2	56.1	58.8	54.9	58.6	54.4	59.1	54.5	58.0	53.7	58.6	54.4
59.9	54.9	58.8	54.6	58.0	53.7	58.5	54.0	57.7	53.5	58.5	54.0
59.2	54.9	58.1	53.8	57.6	53.2	58.0	53.8	57.2	53.2	57.7	53.5
58.5	54.3	57.6	53.4	57.0	52.8	57.6	53.3	56.6	52.8	57.2	53.1
57.5	53.5	56.8	52.7	56.2	52.5	56.9	53.0	56.0	52.3	56.5	52.6
55.5	51.8	55.1	51.4	54.9	51.3	55.4	52.2	55.0	51.5	55.1	51.6
53.9	50.5	53.4	50.4	53.5	50.3	54.8	51.2	53.7	50.6	53.8	50.6
52.2	49.8	52.0	49.8	51.9	49.6	53.4	50.6	52.7	50.2	52.5	50.1
...	...	...	...	...	...	...	...	...	...	...	...
66	66	60	60	70	70	50	50	50	50	59.2	59.2
51.3	51.3	61.1	61.1	60.5	60.5	60.8	60.8	54.4	54.4	...	...
...	24	...	67	...	100	...	0	...	88	...	54

DEC. 1947					JAN. 1948				FEB. 1948				
2		9	18	AVG	12	20	27	AVG	3	10	20		AVG
X_n	X_s	X_n	X_n	X_n	X_n	X_n	X_n	X_n	X_n	X_n	X_n	X_s	X_n
34.8	38.5	34.8	35.3	35.0	34.1	33.0	33.0	33.4	29.5	28.0	24.8	33.4	27.4
37.6	40.9	37.3	38.0	37.6	35.6	34.6	34.7	35.0	32.0	31.2	31.4	35.8	31.5
40.2	43.3	39.7	40.2	40.0	37.4	36.6	36.7	36.9	34.0	33.4	33.2	36.8	33.5
43.3	45.7	42.6	42.4	42.8	39.7	38.7	38.8	39.1	36.1	35.1	34.8	38.2	35.3
45.7	47.4	45.0	44.7	45.1	41.8	40.7	40.9	41.1	38.5	37.4	36.8	39.7	37.6
48.3	48.7	47.1	46.7	47.4	43.6	42.4	43.0	43.0	40.5	39.6	39.0	41.0	39.7
49.8	49.8	49.1	48.3	49.1	45.1	44.0	44.4	44.5	42.0	41.4	40.4	42.0	41.3
51.3	50.6	50.3	49.8	50.5	46.2	45.1	45.5	45.6	43.3	42.5	41.9	42.8	42.6
52.2	51.2	51.4	50.7	51.4	47.4	46.1	46.5	46.7	44.5	43.8	43.2	43.5	43.8
52.8	51.6	52.0	51.5	52.1	48.3	47.1	47.5	47.6	45.5	44.6	44.2	44.4	44.6
53.3	51.8	53.1	52.7	53.0	49.8	48.6	49.0	49.1	47.0	46.3	46.2	45.7	46.5
53.0	51.8	53.1	53.3	53.1	50.5	49.7	50.0	50.1	48.4	47.6	47.2	46.2	47.7
53.0	51.5	52.6	53.7	53.1	51.2	50.4	50.8	50.8	49.0	48.6	48.5	47.0	48.7
3.5	3.5	7.1	6.0	....	2.8	5.3	5.3	....	5.0	6.5	0.4	0.4	....
27	27	3	14	17.7	14	19	−11	10.0	0	10	4	4	15.6
16.7	16.7	17.5	18.6	....	23.0	3.7	−2.7	....	11.1	0.7	24.3	24.3	....
....	....	....	....	....	....	....	....	....	....	....	....	100	....

TABLE 1. GROUND TEMPERATURE—AUG

SERIES X_n—BARE GROUND SERIES

DEPTH BELOW SURFACE (FT)	MARCH 1948								
	2	9	18		25		AVERAGE		6
	X_n	X_n	X_n	X_s	X_n	X_s	X_n	X_s	X_n
1	31.2	31.1	31.4	33.7	31.5	32.4	31.3	33.0	40.
2	31.7	31.7	31.5	34.8	31.5	33.1	31.6	33.9	38.
3	32.4	32.7	32.4	36.2	32.4	34.1	32.5	35.1	36.
4	34.1	34.3	34.0	37.9	33.8	35.4	34.0	36.6	36.
5	36.0	35.7	35.4	39.1	35.1	36.8	35.5	37.9	36.
6	38.1	37.8	37.2	40.4	36.8	37.9	37.9	39.1	38.
7	39.8	39.3	38.5	41.3	38.1	38.7	38.9	40.0	39.
8	40.9	40.8	39.5	42.0	39.4	39.6	40.1	40.8	40.
9	42.1	42.0	40.9	42.9	40.5	40.4	41.4	41.6	41.
10	43.1	42.8	41.8	43.8	41.4	41.1	42.3	42.4	42.
12	45.0	44.4	43.6	45.0	43.1	42.1	44.0	43.5	43.
14	46.3	45.8	44.9	46.0	44.4	43.1	45.3	44.5	45.
16	47.4	46.9	46.0	46.3	45.6	43.5	46.5	44.9	46.
Depth of Snow on Ground in.	8.3	7.4	....	....	0.0	0.0	....	....	0
Mean Air Temp.	24	0	38	38	54	34.0	27.3	27.3	40
Avg Air Temp for Preceding Week	27.1	18.4	21.1	21.1	34.3	34.3	....	....	40
Percent Sunshine	....	....	....	5	....	80	....	66	..

ring arrangement at the end of a 20-ft length of steel rod, ⅜ in. in diameter, and pulled taut. The rod was inserted into the test hole to the required depth, and the test hole was filled with earth and carefully tamped at intervals. Finally, the steel rod was removed.

Fig. 1 shows the thermocouple locations and the ground thermocouple junction box. Thermocouple readings, taken by a manually operated potentiometer, were read weekly for the period from August 1947 through April 1948.

SOD PROVIDES INSULATING EFFECT

Ground temperature readings, depth of snow on the ground, mean air temperature, average air temperature for the preceding week and percent of sunshine are shown in Table 1. The insulating effect of the sod is quite apparent. The average temperature difference between the two thermocouple series for the month of September was 7.6 F at the 1-ft depth and progressively decreased to a 3.6 F temperature difference at the 16-ft depth.

During the period between September 1947 and April 1948 the series X_n thermocouples was more readily affected by the air temperature than was the X_s series. This was undoubtedly due to the insulating effect of the sod on the X_s series. Likewise, for the 1947-1948 season, the frost line depths for the X_n and X_s series were about 3 ft and 1 ft, respectively. This should not be

RATURE—AUGUST 1947 TO APRIL 1948 (*concluded*)

UND IN SERIES X_s—GROUND COVERED WITH SOD

	APRIL 1948								
AGE	6	13		24		30		AVERAGE	
X_s	X_n	X_n	X_s	X_n	X_s	X_n	X_s	X_n	X_s
33.0	40.7	43.5	41.4	53.5	49.0	54.9	50.8	48.2	47.1
33.9	38.4	40.4	40.5	50.0	45.6	54.2	49.7	45.7	45.3
35.1	36.5	39.7	40.0	46.9	43.4	51.6	47.8	43.7	43.7
36.6	36.2	39.0	39.8	44.6	42.0	49.0	45.9	42.2	42.6
37.9	36.9	38.8	39.8	42.9	41.4	46.8	44.6	41.3	41.9
39.1	38.0	39.0	40.2	41.7	40.8	44.8	43.8	40.9	41.6
40.0	39.2	39.5	40.4	41.1	40.8	44.3	43.3	41.0	41.5
40.8	40.2	40.2	41.0	41.1	40.8	43.7	43.0	41.3	41.6
41.6	41.2	40.5	41.4	41.2	41.0	43.4	43.0	41.6	41.8
42.4	42.0	41.5	41.9	41.7	41.4	43.5	43.0	42.2	42.1
43.5	43.7	43.2	43.2	42.8	42.3	44.2	43.4	43.5	43.1
44.5	45.0	44.6	44.2	43.8	43.0	45.0	44.4	44.6	43.9
44.9	46.0	45.7	44.7	45.0	43.6	46.0	45.0	45.7	44.4
....	0	0	0	0	0	0	0	....	....
27.3	40	48	48	68	68	48	48	50	50
....	40.3	43.9	43.9	55.7	55.7	57.3	57.3	....	....
66	....	....	100	....	72	....	0	....	62

interpreted as the maximum frost line depth for this locality; for during the 1947-1948 season, air temperatures below zero degrees were infrequent and temporary when they did occur.

The test data for the X_n series from Table 1 are shown in graph form in Fig. 2. Ground temperatures at the 1-ft level lag approximately three weeks behind the air temperature, but have approximately the same characteristic curve. The temperature at this depth varied from 76.6 F on September 9, 1947 to 24.8 F on February 20, 1948. As temperatures were read weekly, the curves of Fig. 2 do not necessarily represent the maximum or minimum temperatures that occurred during the test period.

Fig. 3 shows the ground temperature variation for the X_s series from August 1947 to December 1947. Although these temperatures are consistently lower due to the insulating effect of the sod, they follow the same general characteristics as the curves of Fig. 2.

Fig. 4 shows the monthly variations in ground temperatures for 1947-1948. Average monthly temperatures were used in plotting the curves and therefore are not the lowest recorded temperatures for the various depths. The variation at the 1-ft depth was from 75 F to 27.4 F and at the 16-ft depth from 53.6 to 45.7 F.

If the ground were to be used as a heat source for a heat pump installation and a continuous heat source temperature of 45 F were required, the curves

indicate that it would be necessary to go down to a depth of 14 ft before this condition could be obtained. Some indication of the time lag for this soil can be obtained from these curves. The maximum effect of the air temperature during August does not occur until November at the 16-ft depth—a time lag

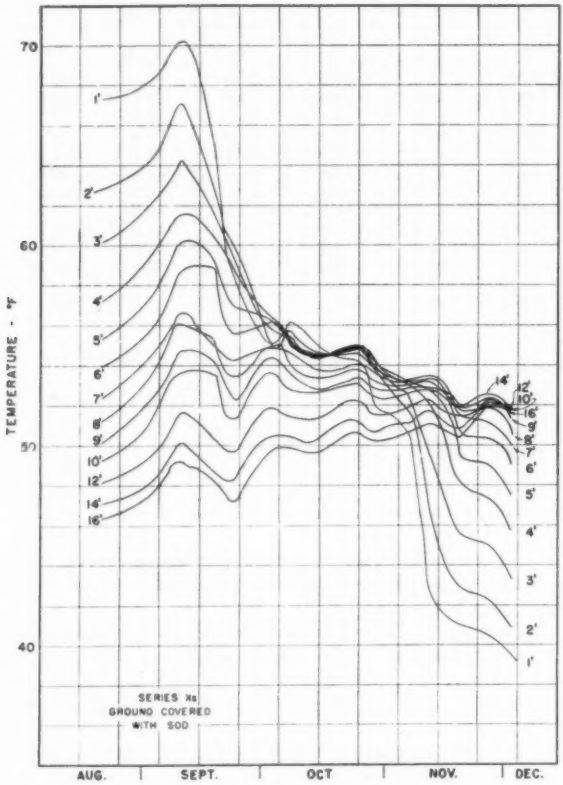
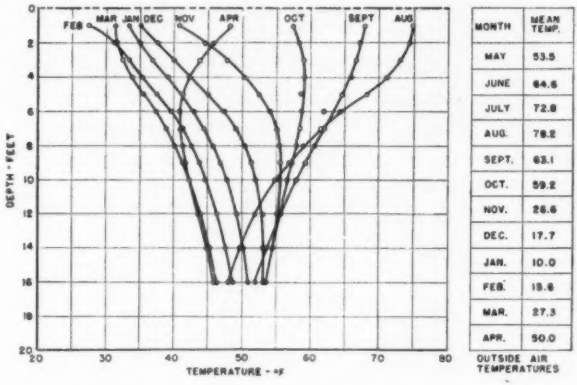


FIG. 3. GROUND TEMPERATURES AT DIFFERENT DEPTHS (AUGUST 1947 TO DECEMBER 1947)

of three months. Likewise, the minimum temperature due to lower air temperatures does not occur until April.

The curves of Fig. 5 show the minimum winter ground temperatures, the maximum summer ground temperatures, and the average ground temperatures at the various depths for 1947-1948. The plotted test points show a reasonably smooth curve. If these curves are extended it will be noted that they tend to approach a common temperature. At the 28-ft depth the difference in tem-



MONTH	MEAN TEMP.
MAY	53.5
JUNE	64.6
JULY	72.8
AUG.	70.2
SEPT.	63.1
OCT.	50.2
NOV.	36.6
DEC.	17.7
JAN.	10.0
FEB.	15.6
MAR.	27.3
APR.	50.0

OUTSIDE AIR TEMPERATURES

FIG. 4. MONTHLY VARIATIONS OF GROUND TEMPERATURES 1947-48

perature between minimum winter and maximum summer is approximately two degrees. The average temperature at this depth is about 50 F. This appears to be both consistent and reasonable as the temperature of the Artesian water supply in this locality is between 48 F and 50 F.

Although the test data included in this paper deal only with one soil, an analysis of the classification of various soils would indicate that for practical purposes moist soils could be included in the following general classifications:

1. *Sandy soils:* soils with 50 percent or more of sand and gravel content (particle size is larger than 0.05 milimeters).
2. *Silt and clay soils:* soils containing less than 50 percent sand and gravel.

The soil referred to in this paper would come under the general classification of silt and clay soils.

TABLE 2. TEST REPORT ON SOIL SAMPLE[a]

	SAMPLE 1	SAMPLE 2
	Mechanical Analysis	*Percent*
Silt	54.6	48.9
Clay	16.4	14.3
Sand	29.0	36.8
	Physical Constants	*Percent*
Moisture, field conditions	23	26.1
Density, field conditions	118.2 lb per cu ft	108.4 lb per cu ft
Density, dry basis	96.1 lb per cu ft	86.0 lb per cu ft
Specific gravity	2.64	2.65
Percolation	5 min	6 min

[a] Tests conducted by Prof. Miles S. Kersten at the Engineering Experiment Station, University of Minnesota.

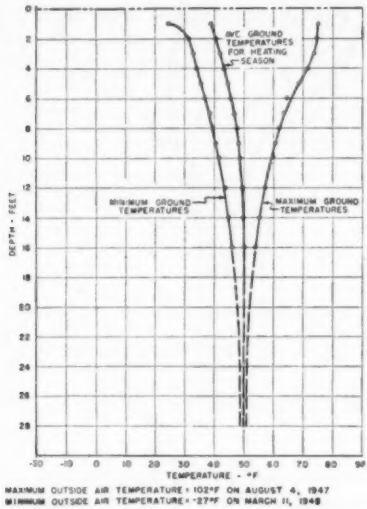

AVG. GROUND TEMPERATURES FOR HEATING SEASON
MINIMUM GROUND TEMPERATURES
MAXIMUM GROUND TEMPERATURES
DEPTH - FEET
TEMPERATURE - °F
MAXIMUM OUTSIDE AIR TEMPERATURE = 102°F ON AUGUST 4, 1947
MINIMUM OUTSIDE AIR TEMPERATURE = -27°F ON MARCH 11, 1948

FIG. 5. DIFFERENCES IN GROUND TEMPERATURES 1947-48

The ground temperatures reported would not be of much practical value without additional information on the soil itself. To obtain this information, both laboratory and field tests were made. Two samples of soil were examined: *Sample 1* taken between a 2- to 3-foot depth; and *Sample 2,* taken between the 1- and 3-ft depth. Results are reported in Table 2.

For the conductivity tests, the average dry density of the two soil samples was used. The soil to be tested was first mixed to the desired moisture content and then compacted into the container at the desired density. The containers

TABLE 3. RESULTS OF CONDUCTIVITY TESTS[a]

TEST NO.	DENSITY LB/CU FT	MOISTURE PERCENT	MEAN TEMPERATURE T	k BTU/SQ FT/HR F/IN.
1	93.0	1.23	70.3	2.23
2	93.0	1.23	40.1	2.16
3	93.0	1.23	24.9	2.14
4	93.0	1.23	−19.9	2.15
5	92.4	16.70	70.1	9.14
6	92.4	16.70	40.1	8.78
7	92.4	16.70	25.1	11.17
8	92.4	16.70	−19.9	10.97
9	92.4	23.0	70.0	11.00

[a] Tests conducted by Prof. Miles S. Kersten at the Engineering Experiment Station, University of Minnesota.

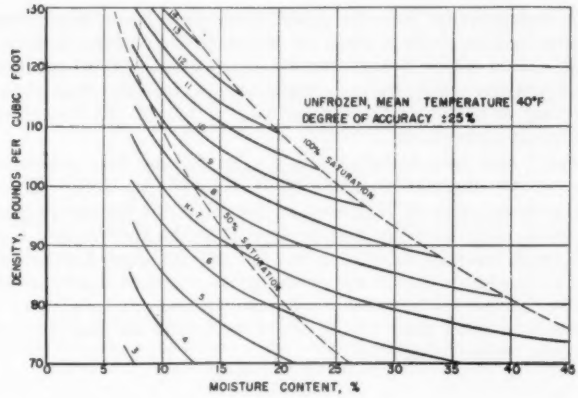


FIG. 6. DIAGRAM OF THERMAL CONDUCTIVITY FOR SILT AND CLAY UNFROZEN SOIL (MEAN TEMPERATURE OF 40 F. DEGREE OF ACCURACY ±25 PERCENT)

were sealed at both bottom and top to prevent loss of moisture. Table 3 gives the results of conductivity tests.

The test results indicate that the thermal conductivity of the soil above the freezing temperature increased only slightly with an increase in mean temperature for constant density. Likewise, for a moisture content up to 16 percent,

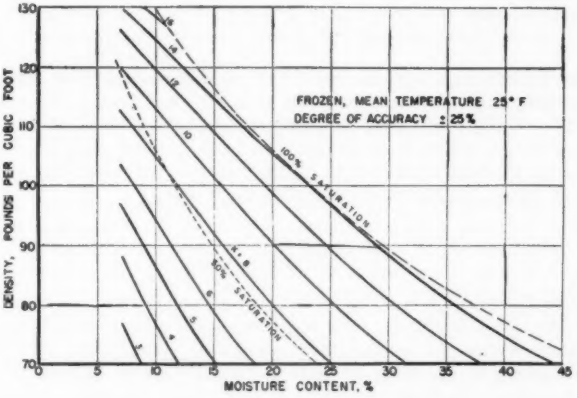


FIG. 7. DIAGRAM OF THERMAL CONDUCTIVITY FOR SILT AND CLAY FROZEN SOIL (MEAN TEMPERATURE 25 F. DEGREE OF ACCURACY ±25 PERCENT)

the thermal conductivity does not vary appreciably in a mean temperature range of from +25 to −20 F. For the low moisture content (2 to 9 percent) the conductivity is lower below freezing than above. With an increase in moisture content the conductivity is higher below freezing than above. It is also interesting to note that the conductivity of frozen soil decreases with a decrease in mean temperature.

Figs. 6 and 7 have been included in this paper because they present extensive information on the thermal conductivity of silt and clay soils. The material is part of the results of an investigation to determine the thermal properties of a variety of soils[1]. The work has been performed by the Engineering Experiment Station of the University of Minnesota for the St. Paul District, Corps of Engineers, U. S. Department of the Army. The curves of Fig. 6 present information on the thermal conductivity of unfrozen silt and clay soils tested at a 40 F mean temperature, while those of Fig. 7 are for frozen silt and clay soils tested at a 25 F mean temperature.

CONCLUSIONS

An analysis of the conductivity test results indicates that the thermal conductivity of the silt and clay soils varies as follows:

1. At a constant moisture content, it increases with an increase in dry density. The rate of increase is fairly constant and is independent of the moisture content.

2. At a constant dry density, it increases with an increase in moisture content.

3. For the unfrozen soil, it increases with an increase in mean temperature.

4. The frozen soil at low moisture content shows very little change; in general, for soils of high moisture content it shows an increase for a decrease in temperature.

5. When changing from unfrozen to frozen soil, the conductivity varies according to the moisture content. For dry soils, it does not change; for soils of low moisture content, it decreases; and with soils of high moisture content, it increases.

Although it was not considered within the scope of this paper to make application to the published mathematical theory underlying heat conduction in the ground, a brief check indicated a close correlation between the calculated results based on test data and the results obtained by mathematical analysis. Additional information on thermal conductivity, density, specific heat, specific gravity and diffusivity of some nineteen different soils is available[2].

ACKNOWLEDGMENTS

This research project was made possible by grants from the Graduate School of the University of Minnesota and the Minneapolis-Honeywell Regulator Co.

The author wishes to acknowledge assistance from John V. Borry and Robert D. Turnacliff, graduate students in 1946-1947, and Marvin N. Lee and John R. Jamieson, graduate students in 1947-1948, for taking of test data,

[1] The Thermal Conductivity of Soils, by Miles S. Kersten, University of Minnesota Engineering Experiment Station. (Presented at the Annual Meeting of the Highway Research Board, Washington, D. C., December, 1948.)

[2] Thermal Properties of Soils, Engineering Experiment Station, *Bulletin 28*, University of Minnesota.

analysis of test data and other helpful suggestions. The author is particularly indebted to Prof. Miles S. Kersten, Civil Engineering Department, for laboratory test results and to Prof. Frank B. Rowley, director of the Engineering Experiment Station, for cooperation in making available testing equipment.

DISCUSSION

R. S. DILL, Washington, D.C. (WRITTEN): The data reported in this paper are interesting and useful indications of what kind of underground temperatures can be expected in the Minnesota climate. In my view, however, the work, for completeness, merits extension to other regions and to more than one season in a single locality.

The paper states that *ground temperatures at the one-foot level, lag approximately three weeks behind the air temperature and have approximately the same characteristic curve.* This undoubtedly was indicated by the data gathered but it is due, I believe, to the period of observation which was, in general, on the order of one week between readings. Data presented in National Bureau of Standards Building Materials and Structures Report No. 103 (Measurements of Heat Losses from Slab Floors, Government Printing Office) show that earth temperatures at the one-foot level follow air temperature changes within a couple of days when the temperatures are read once per day or more often.

Air temperature changes are like waves on a sea; there are long rollers on which there are waves and on which, in turn, there are ripples. Likewise, in air temperature there is the annual cycle which is predictable within limits. Also, there are monthly, weekly, daily and hourly changes which are not predictable with certainty except that night temperatures are expected to be lower than day temperatures. So far as earth temperature at a particular point and time is concerned, one governing factor is the average air temperature for some period preceding the instant of observation. A short, sharp cold snap may not have as much influence as a longer period of moderate cold. This suggests that earth temperatures might be better correlated with degree days, or something similar, rather than air temperatures. Also, the incidence of solar energy is obviously influential and I believe it could well be discussed in a paper on this subject.

At these shallow depths it is doubtful that conditions are materially affected by earth heat. This is indicated by the existence of permafrost or eternal ground-ice in arctic regions. The inference is, therefore, that the ground temperature at shallow depths is governed by heat transfer to and from the surface. The author's attention to the possible influence of sod or vegetable covering is therefore justified, while it is noteworthy that the effect of snow as an insulator is ignored. In fact, the unit of measurement for snow depths is not given, and a reader is in doubt whether the snow attained depths of 7 or 8 in. or 7 or 8 ft. Snow or another insulator on the surface would be expected to make vertical temperatures more nearly the same.

Another effect which the author might consider when space permits is that of the latent heat of the water in the earth. If damp earth is cooled by extracting heat at a constant rate from above to below the freezing point, a plateau would be expected on a temperature-time curve indicating a pause in the temperature drop while the latent heat of fusion is extracted. Fig. 2 indicates that freezing weather commenced about the middle of October and plateaux are evident for several depths shortly thereafter. Early in January, the ice layer reached the 1-ft level and a sharp temperature drop then occurred at that point. This was followed by a rapid rise as the weather warmed and another plateau indicating an absorption of latent heat as the ground thawed.

It should be remembered that earth temperature alone is not sufficient for heat pump coil design. If a coil were installed and heat extracted in the location and

at any depth considered, freezing temperatures around the coil could be expected shortly after or even before the air temperature descended to freezing. The evaporator temperature should not be assumed equal to the earth temperature indicated by the data.

The data given on earth conductivities will answer a long-felt need and the author certainly deserves our thanks for gathering the information which the paper contains on sub-soil conditions.

J. D. KROEKER, Portland, Ore. (WRITTEN): The march of temperatures shown are of value in a general consideration of the thermal problems encountered in both panel heating and ground source heat pump work. The conclusions seem reasonable and are basic for any consideration of thermal properties of the soil.

The project is encouraging in that it indicates that work is being done in a field where so little is actually known, but its greatest use can be in showing the necessity of standardization in the methods and details used in research work along this line. For example, it is possible that the soil classification referred to in the paper is over-simplified. The Casagrande or Army method of classification, used by some other investigators, would seem more definite and flexible. Translation of soil classification to such standard method is suggested to the author in further studies and for consideration of others reporting studies on earth as a heat source. The article did not specifically state whether there was any attempt to maintain the original physical properties of the soil in making laboratory tests for thermal conductivity. This should be done.

Experience with ground coils in some localities has indicated that it is not desirable to bury the coil at the depth of the highest average ground temperature, as was stated in the body of the paper. It has been found that recovery of the soil is too slow when coils are deep. In Oregon where the average winter temperature is about 49 F and the average well water temperature, corresponding to steady state levels, is about 55 F a depth of about 4 ft below the surface seems to be most favorable.

E. P. PALMATIER, Syracuse, N. Y. (WRITTEN): In Fig. 3 was the marked dip in temperature, which is indicated for depths of 16 ft up to about 9 ft about September 20, correlated with any of the known factors affecting these temperatures?

1371

No. 1371

PERMEANCE MEASUREMENT IMPROVED BY SPECIAL CELL

By F. A. Joy* and E. R. Queer**, State College, Pa.

THE PROBLEMS of *humidity* in their relation to comfort, process, or storage are a major concern of air conditioning engineers. Packaging engineers choose envelope materials which retain moisture in bread and exclude it from salt, and the packaging of living spaces is being given more attention. Fundamental to these considerations is the rate of moisture transmission through materials—paper, paint, wood, etc.; the determination of this rate for each material, *i.e.,* its *permeance,* is the first step in solving such problems.

On a basis of unit time, unit area and unit vapor pressure difference, the rate of water vapor transmission through any specific thickness of a material is its *permeance.* Another term, *permeability,* has frequently been used in this application but such use is unfortunate. Permeability should be applied only to a unit thickness of the material, and when so used it becomes a distinguishing characteristic of the substance—one of its physical properties. As an illustration, the permeance of a wood plank is only a small fraction of the permeance of a shaving cut from it, yet their permeabilities are the same. This use of terms is consistent with the established practice in the thermal field where conductance relates to a particular piece of material or combination of such pieces. Conductivity relates to a unit thickness of a material. In electricity a similar use of terms has also been established. Likewise, the vapor *resistance* of a specific piece of material is the reciprocal of its permeance, just as resistance to heat transmission or electricity is the reciprocal of conductance. *Resistivity* to water vapor is the reciprocal of permeability and likewise applies to a unit thickness of the material, just as the same term is applied in the other fields.

Methods of Measurement

Many devices have been used for the measurement of water vapor transmission. The smaller sizes are usually called *Vapor transmission cells* or simply cells and in their simplest form consist of a cup or dish over the top of which the specimen is sealed. If the cup contains water, and is placed in a dehumidified space, the method is called the wet method. If it contains a desiccant, and the

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surrounding space is humidified, it is called the dry method. In either case the vapor transmission through the permeable specimen is determined by periodic weighing of the cell.

Many variations of these basic designs have been developed differing in size of the specimen and the manner of its attachment to the top of the cup. Sealing

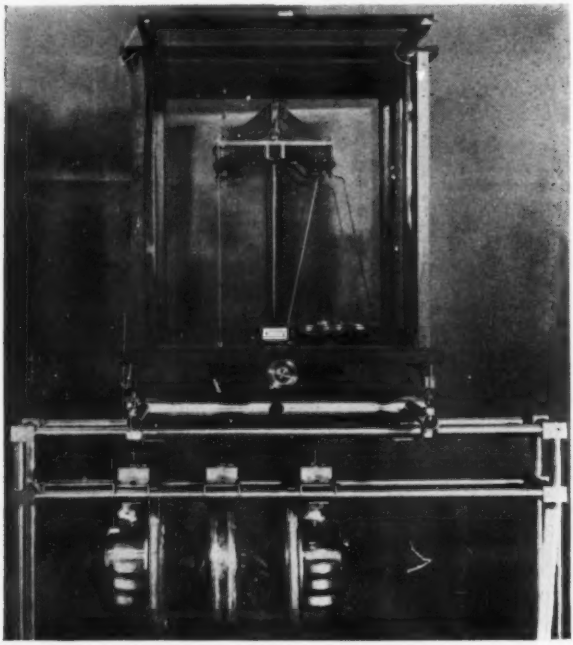

FIG. 1. WATER VAPOR TRANSMISSION CELL—MODEL 1

with waxes is most common but pressure seals are also used. Air circulation over the external surface is usual and it has sometimes been attempted within the cell.

Four tentative standards have been approved by the *American Society for Testing Materials*. D988-48T, also known as the *Tappi Method*, applies to paper, paperboard, and other sheet materials and is essentially a dry method. C214-48T, applicable to vapor seals up to ⅛ in. thick, used in connection with thermal insulation, approves either the wet or dry method, with the warning that the results *should not be compared*.

All such methods are naturally isothermal, the vapor pressure difference across the specimen being due to the difference of relative humidity inside and outside the cell. Other devices have been developed which provide a temperature difference across the specimen. They are applicable to thicker materials

and are naturally of larger dimensions and less simple. They apply especially to the testing of building materials for which a temperature difference is the normal service condition.

An outstanding feature of the many methods in use is the disagreement of their results. This disagreement is especially evident when results obtained by the *wet* and *dry* methods are compared and therefore a statement of the method used is necessary. Accumulating evidence indicates that the rate of water vapor transmission through some specimens depends not only upon the vapor pressure difference, but also upon other factors such as relative humidity and temperature, as well as air movement over the surfaces.

To overcome some of the limitations of the simpler methods and especially to provide a means for testing thicker specimens, a project at the Engineering Experiment Station of The Pennsylvania State College was initiated in 1944.

DOUBLE CUP CELL

At the outset it was decided that the apparatus should combine the good features of both wet and dry methods, using two cups between which the specimen could be sealed. This arrangement, while still requiring constant ambient temperature, eliminates the need for humidity control of the atmosphere surrounding the cell. It greatly reduces the problems of entrance into and necessary work within that space, or the alternative of remote handling of cells for weighing purposes.

On the other hand, the double cup cell presents other weighing problems. No longer is the changing weight of the whole cell a measure of water vapor transfer. In the new arrangement there is no change in weight of the whole cell. Instead, it is necessary to provide within one cup or *chamber* (as hereafter designated) a water pan, and within the other, a desiccant pan.

The loss in weight of the water is the weight of moisture entering the specimen, and the gain in weight of the desiccant is the weight of moisture leaving the specimen[1]. If the specimen is hygroscopic, these weight changes will not at first be equal while the specimen is adsorbing or giving up moisture in an approach to equilibrium, or a *steady state* condition. When a steady state is established, the water loss and the desiccant gain are equal.

The *equality of inflow and outflow rates* as an indicator of a steady state *is fundamental to this test method.* Inequality, on the other hand, has two possible interpretations: (1) a steady state is not reached, or (2) the cell is not tight and moisture is entering or leaving the cell[2]. The opportunity to make two separate determinations of the weight of water vapor transmitted is a unique feature of the two chamber design and possesses marked advantages in the detection of errors in data or malfunctioning of the apparatus.

Accompanying these manifest advantages is the problem of weighing the water and desiccant pans periodically without removing them from their respective chambers. In the first preliminary design (Fig. 1) this problem was solved

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[1] This statement is not quite true during the first hours of a test while the chamber walls are adsorbing or giving out moisture in their approach to an equilibrium with the adjacent atmosphere.

[2] A third possibility is a variation of No. 2, namely: that moisture is condensing on the chamber surface—an eventuality in the *wet* chamber at high relative humidity. A fourth possibility—chemical reaction—is eliminated by constructing the cell of non-corrodible materials.

by suspending each pan (actually a set of 4 pans) from a rod entering the chamber through a rubber bellows. At the top end, each rod was hooked for attachment to an analytical balance movable on suitable rails to allow accurate positioning above each of its loads. The rubber bellows was promptly found unsatisfactory, however, as it reduced the precision of weighing and was not impervious to water vapor. A liquid seal was also tried using kerosene but its buoyancy and evaporation were objectionable.

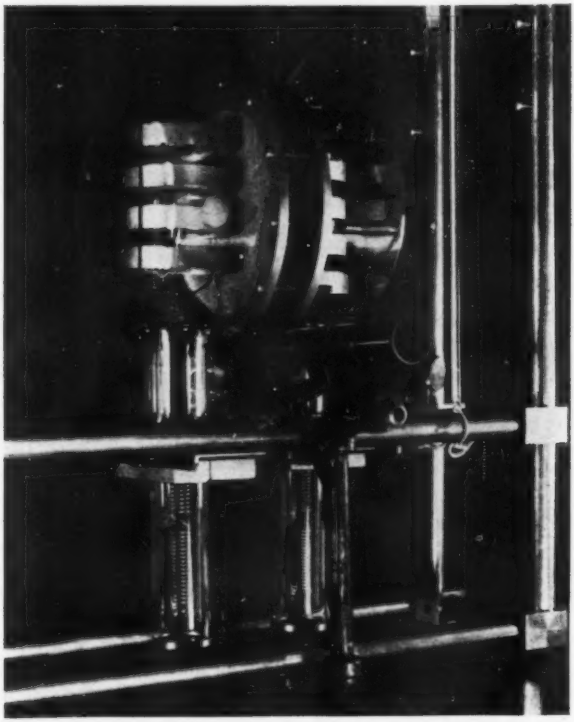

FIG. 2. WATER VAPOR TRANSMISSION CELL—MODEL 2

A feature also attempted in this first model was the periodic weighing of the specimen itself as a further check on its water adsorption. Flexible plastic sleeves attached the specimen ring to each chamber as shown in Fig. 1, but for the same reasons that had discredited the rubber bellows this arrangement proved impracticable and the attempt to weigh the specimen was abandoned in later models.

In the second preliminary design, Fig. 2, a new method of weighing was used. Each set of pans was supported by calibrated iso-elastic springs enclosed within

a glass tube which formed an extension of the chamber, sealed at the top. The helical springs were arranged in pairs, one being wound opposite to the other so that their stretching produced no rotation of the load. This was necessary since the link between load and spring was a graduated scale whose position was sighted with reference to the crosshair in an externally mounted microscope. A change in the scale position could be converted to weight by means of the spring constant

FIG. 3. WATER VAPOR TRANSMISSION CELL—MODEL 3

which was 770 grains per inch. The specimen as shown in Fig. 2 was clamped between the wet and dry chambers with wax used to seal each face to its adjacent chamber and also to seal the rim of the specimen which was otherwise exposed. Some edge leakage was promptly indicated and required correction, but this model was a long step toward a satisfactory design.

The next design, Model 3, is shown in Fig. 3, and in the sketch (with a few adjuncts and separated chambers) in Fig. 4. Twelve such cells have been built and are being used with satisfaction especially in the testing of vapor barriers. The improvements provided in this design include: (1) a steel band enclosing

the specimen, (2) heavier and longer springs, (3) larger specimen area, (4) means to measure relative humidity in each chamber, (5) table support, and refinements.

The chambers are brass castings having glass windows gasketed with neoprene. The specimen is fitted and sealed within a 12 in. diameter steel ring

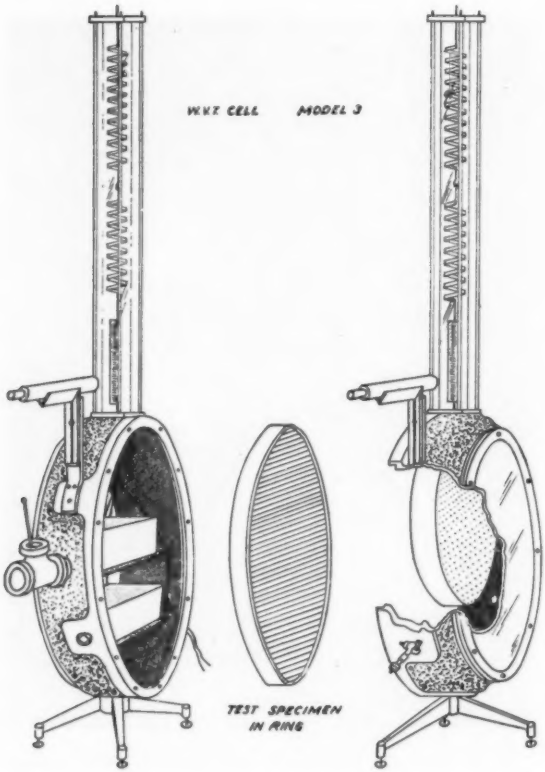

FIG. 4. W.V.T. CELL—MODEL 3 AND TEST SPECIMEN IN RING

which may be of any desired width to accommodate a thick or thin specimen. The ring edges are machined, and the ring rather than the specimen provides the contact surfaces when the test assembly is clamped between the two chambers.

The glass columns enclose springs that combine a high initial loading with a satisfactory spring constant, 1952 grains per in., and a load carrying capacity of 3½ lb. The importance of high load capacity applies especially to the dry side and will be discussed later.

Through the cell window in Fig. 3 is seen the desiccant container, made of aluminum with perforated sides. Fig. 4 shows the water containers—two shallow glass dishes stacked one above the other to provide a water area of 18 sq in. Glass permits the use of saturated salt solutions, the choice of salt providing a means of adjusting in a limited range the wet side relative humidity. An adjustable tripod and an adjustable microscope holder for each chamber are provided.

DETERMINING VAPOR PRESSURE*

Means for determining the vapor pressure in each chamber are important additions shown in Fig. 4. At the rear is fitted an electric hygrometer element and at the front is a dew point cup built into the up-facing branch of a 2 in. pipe tee. The cup is supported in an adapter made of ½ in. plate lucite, a material which provides the necessary heat isolation for the cup and good vapor integrity for the chamber. In the front of the tee, a window is fitted for observation of the polished cylindrical cup. Plate lucite is again used for easy machineability and fitting into the pipe threads.

The dry side dew point cup is usually cooled by dry ice in an alcohol bath while the wet side usually requires only the insertion of a chilled metal rod into the water it contains. The bath is allowed to remain in each cup between observations but both cups are stoppered. Avoidance of evaporation from the wet side cup is especially important since its lowered temperature is apt to produce a continued condensation within the cell.

In the set-up of a test it is desirable to test the tightness of the closure to air pressure. A stop cock in each chamber and suitable rubber tube fittings provide a means for raising the pressure (with a hand pump) in both chambers simultaneously to about 1 in. Hg. Experience indicates that absolute tightness is not important but gross leakage should be avoided.

This design has been used with satisfaction in the tests of many specimens, such as sheet vapor barriers for building construction and paint coatings, whose permeance (under the test conditions) was less than 5 grains per (square foot) (hour) (inch of mercury), or 5 *Perms*.

If the specimen has a higher transmission rate, the vapor resistance of the air paths from the water surfaces to the test panel on the wet side and from the panel to the active desiccant on the dry side, constitute too large a portion of

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* The weight of water vapor transferred through a particular piece of material is undoubtedly proportional to its area, and to the time duration of the transfer, but it is frequently not proportional to the vapor pressure difference across the panel. Nevertheless, the practice of relating vapor transfer to vapor pressure difference is well established and useful. When expressing permeance in *grains per (square foot) (hour) (inch of mercury)* it is important to note that the number so obtained is a measure of permeance in the particular situation. If the situation involves a pressure difference not 1 in. Hg, the description of the measurement does not imply that this number of grains would be the weight of vapor transferred if the pressure difference were 1 in. Hg. The validity of such an implication would require that the permeance be constant which, for many materials, it is not.

In any event, this unit *one grain per (square foot) (hour) (inch of mercury)*, although convenient to use, is a bit unwieldy to write or talk about, and it is therefore proposed that the brief designation *Perm* be the name applied to it. This name has been used in the text.

the whole resistance. As a result the vapor pressure is not uniform over the whole surface of the specimen on either side, and its measurement is uncertain.

Another limitation, significant only in the testing of a specimen of higher permeance, is one which is characteristic of the dry method. Since the vapor pressure in the dry chamber rises steadily as the desiccant adsorbs water, a steady state for hygroscopic specimens is technically unattainable. This rise, however, is minimized by the use of a large effective quantity of desiccant.

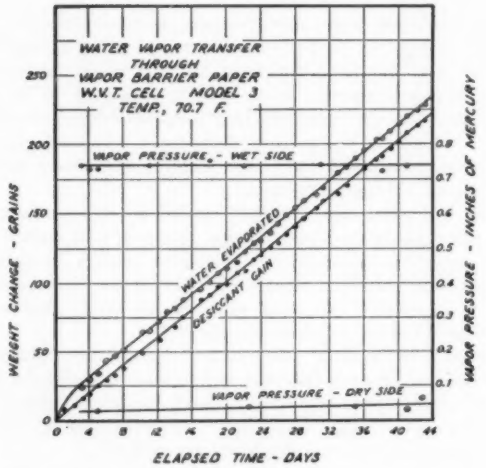

FIG. 5. WATER VAPOR TRANSFER THROUGH VAPOR BARRIER PAPER IN W.V.T. CELL—MODEL 3 (TEMPERATURE 70.7 F)

Moreover, the slight rise that occurs in the testing of specimens of low permeance such as vapor barriers or painted panels may properly be ignored.

These facts are illustrated in the following examples which are actual tests made with the Model 3 cell. In Fig. 5 is shown the plot of data taken in the test of a vapor barrier *paper*—a duplex paper with an asphalt center. The daily scale readings (in thousandths of an inch) have been converted to weights using the spring calibration, 1952 grains per inch of deflection, and the results plotted as *water evaporated* and *desiccant gain*. The points lie very close to straight lines, the departure being rarely in excess of two grains which is the practical limit of scale readability.

At the start, the divergence of these lines indicates that an amount of about 12 grains of water vapor has been adsorbed by the paper and the water chamber walls in their approach to a steady state. The amount given up by the dry chamber walls and the adjacent side of the specimen (and picked up by the desiccant) is either too slight to be indicated or has already occurred before the first reading. As drawn, these lines are not quite parallel, but the divergence is

only two grains in 44 days. The slowly increasing relative humidity in the dry chamber is a very slight factor but most of this two grain difference may be considered leakage. In either case, its magnitude is negligible.

The vapor pressure in each chamber was obtained by the dew point method, since the relative humidity was outside the satisfactory range of available electric hygrometer elements. The plots of these pressures also are shown in Fig. 5, and straight lines are drawn through them. On the wet side, the vapor pressure is deemed constant at 0.735 in. Hg which corresponds to 97 percent R.H. (relative humidity). On the dry side, the vapor pressure is not quite constant, the slight rise corresponding to a rise in relative humidity from 3.3 percent at the start to 6.6 percent R.H. at the finish.

This 3.3 percent increment produces a 3.5 percent reduction in the vapor pressure difference across the specimen, but its effect on the vapor transfer rate is not discernible. The choice of straight lines to represent the weight changes in effect averages the rates of transfer at the start and finish of the test, and it is therefore appropriate to use the average vapor pressure difference for the period, namely, $0.735 - 0.037 = 0.698$ in. Hg.

The permeance of the specimen is computed by solving the following equation:

Weight of vapor transmitted = permeance × Area × Time × Vapor pressure difference

or

$$\text{permeance} = \frac{W}{tA\,\Delta p} \quad \ldots\ldots\ldots\ldots \quad (1)$$

where

W = the weight, in grains, of the vapor transmitted
t = the time, in hours, required for the above transmission
A = the area of the specimen in square feet
Δp = the difference in vapor pressure, in inches of mercury, across the specimen

in this case,

$$\text{Permeance} = \frac{223}{(44 \times 24) \times 0.726 \times 0.698} = 0.417 \textit{ Perms}$$

To demonstrate the effects of a more substantial rise in the dry side relative humidity during the progress of a test, the results of another test in the Model 3 cell are shown in Fig. 6. In this case, the specimen (a pine wood panel 1.375 in. thick) was highly hygroscopic, and although the test was continued for 80 days, a steady state was never reached. In fact, as already noted in these circumstances, a steady state is unattainable.

With the scales shown, the plotted points of the data are everywhere touched by the curves, and are therefore omitted from the graphs. The wide and continued divergence of the curves—*water evaporated* and *desiccant gain*—indicates clearly the increasing adsorption of vapor in the wood amounting to about 400 grains at the finish[3]. The changing curvature of each line is the result of several variations, but primarily of the changing relative humidities in both cell

[3] The possibility of leakage out of the cell is not overlooked, but has been substantially eliminated by twice checking the vapor-tight integrity of the cell.

chambers. These changing conditions are indicated by the curves at the top of Fig. 6, the wet side relative humidity rising from 87 percent to 96 percent R.H. accompanying the dry side rise from 2 percent to 34 percent R.H. As a result, the vapor pressure difference across the specimen diminished 27 percent during the progress of the test, yet the rate of transfer was well maintained.

The explanation of this paradox lies of course in the fact that wood permeability increases greatly as relative humidity is increased. The computations of the permeance of this panel are based upon the day to day conditions and the results plotted to give the curve at the bottom of Fig. 6, the dry side weights

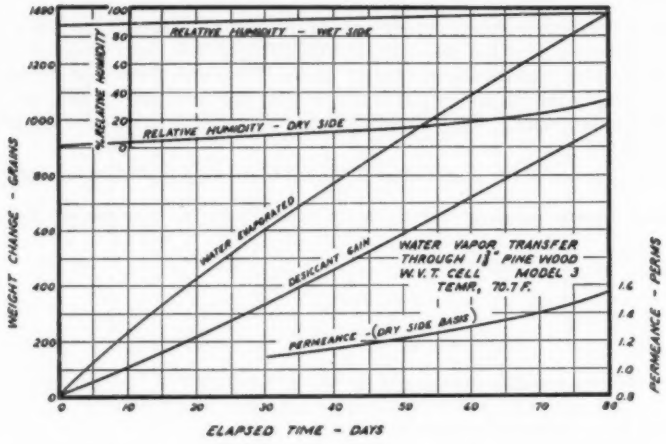
RELATIVE HUMIDITY - WET SIDE
RELATIVE HUMIDITY - DRY SIDE
% RELATIVE HUMIDITY
WEIGHT CHANGE - GRAINS
WATER EVAPORATED
DESICCANT GAIN
WATER VAPOR TRANSFER THROUGH 1⅜" PINE WOOD W.V.T. CELL MODEL 3 TEMP. 70.7 F.
PERMEANCE - (DRY SIDE BASIS)
PERMEANCE - PERMS
ELAPSED TIME - DAYS

FIG. 6. WATER VAPOR TRANSFER THROUGH 1⅜ IN. PINE WOOD IN W.V.T. CELL—MODEL 3 (TEMPERATURE 70.7 F)

being alone considered. From this curve it is seen that the permeance increased 40 percent during the latter 50 days of the test and was rising steeply when the test was terminated.

The permeance thus determined, in an absence of steady state, must be considered as an approximation although the error is probably not great. The test illustrates the difficulty shared in greater or lesser degree by all methods using open desiccants as the vapor sink.

Another difficulty shared in some degree by other methods, both dry and wet, has already been mentioned, namely, the uncertain vapor resistance of the air paths within the cell. However, with these limitations in mind, Model 3 is recommended in the testing of specimens whose permeance is low, if the high relative humidity on one face and the low relative humidity on the other face are acceptable conditions.

In 1948 a project was undertaken for the Office of Technical Services, U.S. Department of Commerce, which included studies of the variation in the permeability of wood with relative humidity and temperature. Tests were required

in a wide range of humidity, sometimes with the high relative humidity on the side of lower vapor pressure and vice versa. These requirements necessitated a more flexible design in which both temperature and relative humidity could be set and maintained in each chamber at any chosen value. To provide these features the controlled humidity duplex cell, Model 4, shown in section in Fig. 7 and in Fig. 8, was constructed.

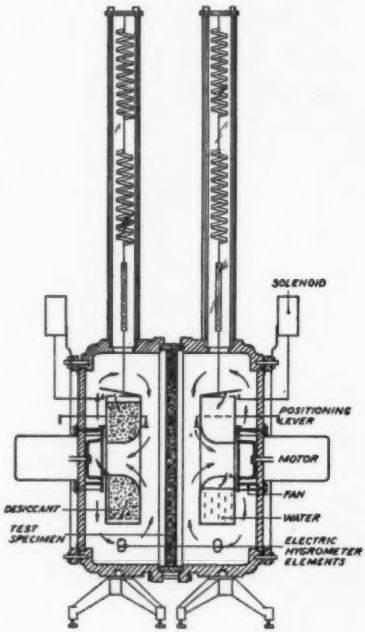

FIG. 7. CONTROLLED HUMIDITY DUPLEX CELL—MODEL 4

Temperature control in each chamber is accomplished by enclosing the complete cell in a box made of structural insulating board divided through the middle by a vertical partition pierced only by the 12 in. ring which holds the specimen. The temperature in each half of the box is thermostatically controlled and a circulating fan provides the turbulence necessary for heat transfer. The warm half is heated by electric heating elements while the cold half is cooled by admitting air from a duct in which 5 F air is continuously circulated at slightly elevated pressure. The 12 in. ring holding the specimen is a band of polyethylene ½ in. thick whose high vapor resistance and low thermal conductance make it suitable as a connection between the warm and cold chambers. In Fig. 8 two of these rings, one holding a specimen, can be seen in the right fore-

ground. In the background are three of the insulated boxes enclosing Model 4 cells in operation.

Within each cell chamber a small 27 cfm fan is fitted, Fig. 7, its four-blade scroll serving to distribute the discharge air fairly uniformly around the peri-

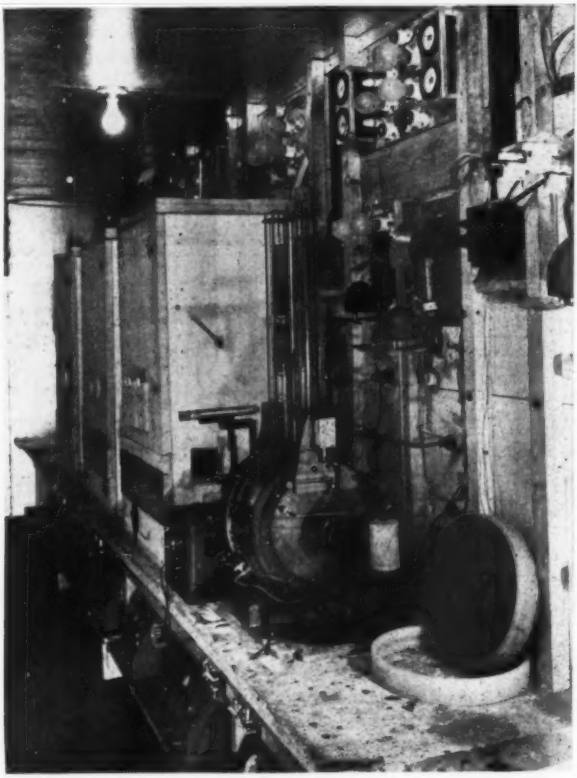

FIG. 8. CONTROLLED HUMIDITY DUPLEX CELL—MODEL 4 WITH CONTROLLED TEMPERATURE ENCLOSURES IN TEST ROOM

phery of the chamber. The driving motor is mounted externally with a seal provided for its entering shaft. The air turbulence set up by this fan serves several purposes, among which is the transfer of heat to or from the panel surface. The air flows radially over the specimen from the rim inward toward the center at increasing velocity in a path guided by the contour of the axially positioned desiccant (or water) container, and returns to the fan suction through a

2 in. hole in the doughnut-like container. The panel surface temperature, indicated by thermocouples, is satisfactorily uniform and constant after the heat balance of the system is established.

The relative humidity in each chamber is controlled at a chosen value. An electric-hygrometer sensing element installed within the chamber is connected to an electronic humidity controller. A second sensing element is also installed for indication only, and each provides a check on the other. These elements are extremely sensitive and indispensable in this design. Their accuracy, of course, is the accuracy of their calibration which is frequently checked in a water jacketed chamber constructed for the purpose. With these provisions, the dew point cup is omitted from this model as a matter of convenience.

The water and desiccant containers are of special design, involving primarily a vapor tight can with two openings through which air may be drawn by the fan when a valve in the entrance port at the top is opened. This valve is lifted by an external solenoid which is energized through a relay of the electronic humidity controller, the mechanical connection being a simple arrangement of levers with the entrance shaft moving only in rotation. Each container is normally positioned adjacent to the fan and centered on its axis by a manual bellcrank whose entering shaft moves both in rotation and translation. The containers are released from all restraint for daily weighing.

The internal features of the desiccant can are as shown at the left of Fig. 7, the air path being downward through the bed into a small plenum from which a pipe leads to the mouth of the shaped orifice hole in the container. The very dry air entering the fan suction opening is mixed in the fan before it is returned to the chamber. This drying is intermittent, being stopped by the closed valve and the effective diffusion resistance of the small outlet tube, but the cycles of valve action are short and the cyclic variation in relative humidity rarely exceeds 1 percent total.

The water can contains features only suggested in the diagram. A vertical partition in the top is located in a plane parallel to the axis of the can and 1 in. off center to clear the entrance port. Through this port the air passes down and around the can, over the water surface and up to an outlet tube which it enters near the top of the can. This tube, like that in the desiccant can, is important as a block to diffusion when the valve is closed but for simplicity is not shown in Fig. 7. These arrangements permit a large storage of very humid air above the water and in effect allow continuous evaporation although the vapor is withdrawn intermittently as required.

Among the desirable features of this cell is positive air movement over the specimen surface. Its velocity is variable and subject to the location of the specimen surface, but ordinarily approaches a maximum of 350 fpm. Although means are at hand for providing any velocity desired, the available evidence suggests that the objective of uniform vapor pressure over the surface is attained by either velocity or turbulence. and that (for a vapor transfer up to 19 grains per hour, the maximum thus far tested), this design fulfills the requirement satisfactorily.

In respect to uniformity of temperature it is evident that a given tolerance and a given air velocity place a limit upon the temperature difference across the panel depending on its thermal conductance. In any event, the tolerance of temperature is a tolerance not of vapor pressure, but of relative humidity, and

requires scrutiny only if the permeance of the specimen is sensitive to relative humidity.

Three cells of this design were built and operated for six months. Good results have been obtained in a wide range of conditions. Mean test temperatures have ranged from 50 F to 110 F with a temperature difference as high as 23 F between panel surfaces. Relative humidities on the water side have been set and controlled at values ranging from 30 percent to 93 percent while operation of the desiccant side has included humidity values from 15 percent to 77 percent. The tested vapor transfer rates have included values from 0.06 to 19

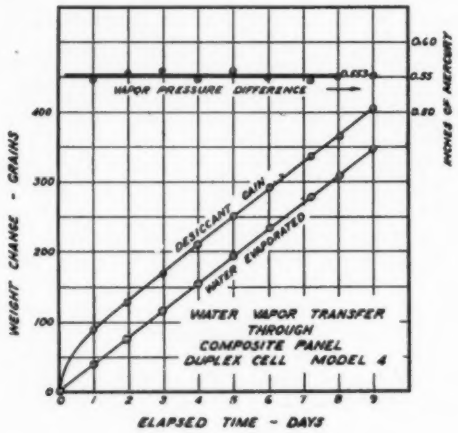

FIG. 9. WATER VAPOR TRANSFER THROUGH COMPOSITE PANEL IN DUPLEX CELL—MODEL 4

grains per hour, and the tested specimens ranged in permeance from 0.34 to 120 *Perms*.

Two actual tests are cited to indicate the nature of data and the significance of results obtained with the Model 4 cell. These tests were made on a composite panel of pine wood and insulating board, the wood thickness being ⅛ in. and the insulating board about ¼ in. They were run in connection with a study of the propriety of adding the vapor resistances of these materials. In the first test the conditions on the water side were 97.6 F, 49.2 percent R.H., and on the desiccant side 73.0 F, 40.6 percent R.H., with a vapor pressure difference of 0.553 in. Hg, the panel being placed with the wood on the water side (at 97.6 F).

In Fig. 9 the scale readings have been converted to weight change (grains) and plotted as *water evaporated* and *desiccant gain*. Readings were taken daily. During the first four of five days, which were required to reach a steady state, the behavior is easily traced as the panel gave up some of its moisture content, having previously been exposed to a higher relative humidity.

In the following four days, the plotted points lie almost exactly in parallel straight lines, the indication of a steady state. From the electric hygrometer

readings, the vapor pressure difference across the panel has been computed and likewise plotted. The stagger of the plotted points, amounting to 1.3 percent above and 1.3 percent below the average, 0.553 in. Hg, does not indicate a shift of conditions from day to day. Rather it is indicative of the band within which the vapor pressure difference was controlled in cycles whose length was not over one minute. The permeance of the panel is obtained as in the foregoing illustration by substituting the test data in Equation 1:

$$\text{permeance} = \frac{W}{tA\,\Delta p}$$

In this case,

$$\text{permeance} = \frac{155}{96 \times 0.66^{*} \times 0.553} = 4.4 \textit{ Perms}$$

A second test on the same panel deserves mention because it illustrates the inconstancy of the permeance of some materials (in this case, wood) and the importance of choosing test conditions which are typical of the service expected of the material.

The conditions of this second test were exactly the same as the first, but the composite panel was merely turned around, so that the wood was on the dessicant side and the insulating board on the water side. The data and calculations are omitted but the permeance was found to be 13.2 *Perms,* which is three times what it was in the other position.

The explanation of this large difference is not hard to find when consideration is given to the relative humidity at the interface between the two materials. Calculations indicate that it was 22 percent in the first test, and 90 percent in the second. In the latter case, the wood permeance which varies widely in response to relative humidity, was greatly increased and accounted for the three-fold rise in the permeance of the combination.

It is significant to note that in tests by some other methods this important difference would have been entirely absent or too small to attract attention.

The conclusion is therefore reached that this test method possesses distinct merit, and that both Model 3 and Model 4 have important uses. Model 3, being simpler and requiring only a temperature controlled room for its operation, is the logical cell for testing materials of lower permeance which are known to be insensitive to relative humidity. The application of Model 4 is limited only by its heat transfer capacity and by the adequacy of an 11¼ in.[4] specimen as a true sample of the material tested.

ACKNOWLEDGMENTS

The original development of this method of test was carried out with the joint sponsorship of the Armstrong Cork Co. and the Engineering Experiment Station of The Pennsylvania State College under the direction of Prof. K. J. DeJuhasz. Valuable assistance was rendered by E. Claxton, Dr. H. A. Robinson, C. C.

* See footnote 4.

[4] The specimen diameter is nominally 12 in. when fitted in a steel ring, and is nominally 11¼ in. when fitted in the thicker polyethylene ring. The waxing technique allows variations from the nominal.

Schrader of the Armstrong Cork Co. Research Laboratories, and Prof. F. G. Hechler, Director of the Engineering Experiment Station. F. D. Yeaple, a graduate student at the College, built the first Model 3 cells and used them in the study of water vapor transmission through holes and slots.

The Office of Technical Services, U.S. Department of Commerce, sponsored the project for which the first Model 4 cells were built.

DISCUSSION

DR. F. T. CARSON, Washington, D. C. (WRITTEN): The apparatus appears to be very well adapted to the sort of uses made of it, and seems rather flexible in use. I do not know whether the authors meant to imply a novel method of weighing (page 380). The use of calibrated springs in a transparent enclosure is frequently used in determining moisture-content isotherms, and this technique was applied to a permeability cell by workers at the Bell Telephone Laboratories several years ago (*Industrial and Engineering Chemistry*, 28, 1255, 1936).

The discussion about definitions and the use of *permeance* and *permeability* is logical, but there seem to be many ideas about how to make a limited number of words go around for all the shades of meaning required. *Permeability* defined so as to characterize a physical property of a substance is of little use to those concerned with materials like paper and textiles that are made and used in fixed thicknesses (not subdivided in use). Permeability as a *piece property* for such materials would have more appeal. Furthermore, the suggested definition involves the nature of the fluid as well as of the substance. It is not independent of the permeating fluid, and would be in conflict with the definitions currently advocated by the *Geophysical Union* and the *American Standards Association*. I am not suggesting how the difficulty can be resolved. The subject is too involved. Prospects for universal agreement on nomenclature are not good.

The name *perm* is an old friend in a new dress. Some years ago when H. D. Hubbard was at the National Bureau of Standards he suggested *perm* as the name for the fundamental cgs unit of permeability to gases (National Bureau of Standards, Research Paper 682, 1934) but it did not find support. I believe it has since been suggested for some other unit. There must be several dozen *permeability* units in use in various fields.

The authors mention a project in 1948 for the Office of Technical Services, U. S. Department of Commerce. I wonder if the report of this work is available (no reference is given). It would be interesting to know what kind of results were obtained under the conditions mentioned (relative vapor pressure and vapor pressure difference opposing each other).

The two-sided effect indicated on page 391 which yields two values on a unit pressure-difference basis (4.4 and 13.2 *perms*) for the same specimen, illustrates the difficulty of expressing permeability data in this manner. It is possible that this two-sided behavior might not have been in evidence if the high-pressure side had not been above 70 or 80 percent relative humidity.

R. S. DILL, Washington, D. C. (WRITTEN): The authors are obviously to be commended on the preparation of an exceptionally comprehensible paper describing some well designed equipment which they have evolved and which should have considerable practical use. The condensation problem in buildings is now becoming generally recognized and understood, and vapor barriers are one means of alleviating it. Devices such as those described should be very useful both for solving problems and settling arguments.

Those interested, however, should remember that the use of vapor barriers, even those with their properties known by means of these instruments, does not entirely solve the condensation problem. In the first place, no matter how well a house is provided with vapor barriers, there will yet be a limit on the tolerable humidity within it because sufficient humidity will result in condensation on windows, doors or their frames and even on the interior surfaces of walls. In the second place, the installation problem always exists. A house vapor seal may be faulty due to design and it is subject to damage by artisans, especially plumbers and electricians.

Questions for which answers may be found by these instruments are: How uniform is vapor barrier material as marketed? Is the permeance of one square foot at all representative of that of another square foot of the same roll? What differences between rolls must be expected? The data contained in National Bureau of Standards, *BMS Report No. 63 (Moisture Condensation in Building Walls)* indicate some considerable variations.

I am in sympathy with the units of measurement proposed by the authors but these units may be acceptable only with difficulty by numerous interested parties.

F. M. GAVAN* and S. W. EBY*, Lancaster, Pa. (WRITTEN): This paper is of particular interest to our company since the transfer of moisture through insulating materials has long been a problem in the industry. The work reported on was sponsored by the company at The Pennsylvania State College for the purpose of making a study of this effect and devising apparatus which would give reproducible measurements.

When six model 3 cells had been built, three were delivered to our research laboratories at Lancaster and a series of cooperative tests was begun in April 1948. This series was designed to obtain data on vapor barrier assemblies of general interest and to determine the agreement which might be obtained when the same samples were tested by two different laboratories. Samples were prepared at both the College and our laboratories. Each laboratory tested its own samples and those prepared by the other.

Although this work has not been completed, the data obtained to date (see Table A) indicate that the agreement between the two laboratories on the same sample is good, and that the method of test differentiates between barriers of varying efficiencies. Where good agreement is not shown (as in the case of our sample C) it can generally be explained by the difficulty in making two assemblies exactly alike. The relative disagreement on sample B may be ascribed to the behavior of the coating in curing. This series will be completed and a statistical analysis is contemplated.

The practical advantages of this type of vapor transmission cell over the widely used *A.S.T.M.* cups are apparent. From the limited amount of work which has been done with both methods on the same material, it appears that the former provides a more reliable test and develops a greater vapor pressure differential across the specimen, producing a greater rate of transfer of moisture. In one test using the new method it was found that the rate of transfer was about three times greater than that obtained with the *A.S.T.M.* wet cup method. It is indicated by a considerable amount of testing that the cell described in this paper duplicates results with greater consistency than the *A.S.T.M.* cup methods.

The work with these cells is by no means finished. Several projects need to be carried to completion to reduce operational difficulties which may now be causes for variation. One of these is to find a standard for calibration, such as a substance of constant permeance, to insure uniform techniques among various operators and laboratories. An improvement now being investigated is a method of indicating electrically the elongation of the springs. Other projects contemplated would utilize the versatility

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TABLE A. PERMEANCE OF VARIOUS VAPOR BARRIERS[a]

SAMPLE	DESCRIPTION	PERMEANCE—GRAINS/(sq ft) (hr) (in. of Hg)			
		PENN STATE SAMPLES		ARMSTRONG SAMPLES	
		Penn State Values	Armstrong Values	Penn State Values	Armstrong Values
A	25/32 in. impregnated insulating sheathing	17.31	16.3		
B	1 in. corkboard—Portland cement plaster	3.59	2.75		
C	1 in. corkboard—asphalt emulsion finish No. 1	0.73	1.17	0.08	
D	Same as (C) with paint No. 1	0.17			
E	Same as (C) with paint No. 2	0.17	0.22		
F	1 in. corkboard—asphalt sealer	0.12		0.08 X[b] 0.03 Y	0.13 X 0.10 Y
G	1 in. corkboard—asphalt emulsion finish No. 2	0.41	0.44		
H	Same as (G) with paint No. 1	0.37	0.37		
I	Same as (G) with paint No. 3	0.13	0.15		

[a] All tests made at 70 F with water on wet side and silica gel on dry side.

[b] X and Y are duplicate samples.

Note: Since a good vapor barrier has been defined (*HHFA* Bulletin No. 5—July 1948) as one having a transmission rate of less than 1.0 perm, it should be noted that these are typical data and not necessarily absolute for the assemblies tested. For instance, it is probable that there is very little practical difference between samples G and H.

of these cells in investigating effects ordinarily allied with water vapor transfer, such as temperature differences, varying vapor pressure differentials, and air flow.

In conclusion it should be noted that this work has apparently resulted in the development of apparatus and techniques which will be of great use and value to all who are engaged in improvement of the design and installation of insulation.

F. B. ROWLEY, Minneapolis, Minn. (WRITTEN): The accurate measurement of the vapor transmitting qualities of a given material often requires very precisely controlled test conditions and delicate measuring devices. This is particularly true for the better types of vapor barriers. In this paper, the authors have described a test apparatus and procedure developed by them, which appears to meet the requirements for a satisfactory test method. I want to compliment the authors upon the carefully worked out design and the principles which they have taken into consideration in developing the apparatus.

This discussion does not apply to the test apparatus, but rather to some of the difficulties involved in the satisfactory rating of vapor barriers as pointed out by the authors in the course of their discussion of test methods and test results.

As stated in this paper, many test methods have been devised and used for the rating of vapor barriers, and there are amazing differences in the ratings as determined for certain barriers by the different methods. In the case pointed out by the

authors, there is a variation of 4.4 *perms* to 13.2 *perms* or 200 percent variation in results based on the lower values. This was obtained by merely shifting the position of two materials in a composite structure. Differences of this magnitude as obtained by different test methods are common and are not due so much to the test apparatus as they are to the omission of certain important elements which affect the permeability rating of certain material. The effect of relative humidity on hygroscopic materials as mentioned by the authors in the last part of the paper has been too often overlooked.

Vapor permeability through materials is affected by several factors. Some of these are the surface resistance of the material to passage of vapor, the internal resistance of

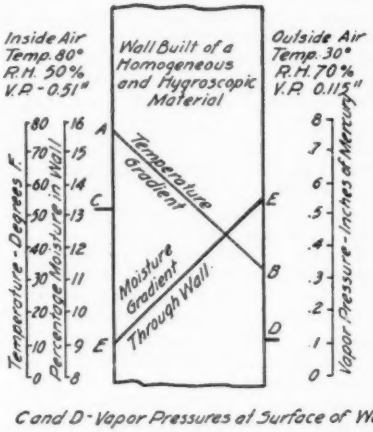

FIG. A. CONTRAST BETWEEN MOISTURE AND TEMPERATURE GRADIENTS THROUGH HOMOGENEOUS AND HYGROSCOPIC WALL

the material to vapor, the temperature of the material, the state of the moisture in passing through the material, and the hygroscopic properties of the material.

When the vapor is transferred to or from the surface of the material, assuming that the surface temperatures are such that no condensation takes place, the rate is affected by the velocity over the surface as well as by vapor pressure difference. When the vapor enters the material, several things may happen. For permeable materials which are non-hygroscopic and whose temperatures are above the dew-point temperature of the vapor all the way through the material, the vapor may pass through and, in general, the rate may be assumed to be directly proportional to the vapor pressure difference on the two sides of the material and inversely proportional to the vapor resistance of the material. If, on the other hand, the material is hygroscopic, some of this vapor may be absorbed by the material, and this absorption may increase or decrease the vapor transmittances depending upon conditions. Furthermore, if the temperature of the material at any point is below the dew-point temperature of the vapor, condensation may take place and the rate may be changed.

If a permeable material which is non-hygroscopic is tested under conditions which do not cause condensation, the permeability appears to be directly proportional to vapor pressure difference, and the rating of the material becomes a rather simple matter.

If the material is to be used under temperature conditions which will cause condensation, the rating as determined by non-condensing conditions may not apply.

If the material is hygroscopic, it will absorb moisture directly from the vapor, and, in general, the amount of water absorbed by hygroscopic materials is proportional to the relative humidity of the vapor in contact with them, but not to the absolute humidity or vapor pressure. This property is the one which appears to be causing the greatest amount of variation between different test methods.

In discussing this principle in a paper before the *Society of Sigma Xi* in 1938, I used Fig. A, as it illustrates very clearly the principle involved. This represents a wall built of an impermeable, homogeneous, hygroscopic material. The left surface of the wall is exposed to air at 80 F and 50 percent R.H., and the right side is exposed to air at 30 F and 70 percent R.H. On the left side there are scales for temperature and percentage of moisture within the wall, and on the right side there is a scale for the vapor pressure of the moisture on each side of the wall.

The line AB represents the temperature gradient through the wall as established by air temperatures on the two sides of the wall. From data taken from a *Forest Products Laboratory* bulletin, the moisture equilibrium content of wood in contact with water vapor of 50 and 70 percent R.H. is 9 and 13.5 percent, respectively. Thus, if the hygroscopic properties of this material were similar to those for wood, the moisture gradient through the material would be represented by the line EE. The vapor pressure over the wall is represented by the points CD. Since the wall is impermeable to vapor, there is no such line in the wall.

Under the conditions shown in this diagram, it is evident that if the material were at all permeable vapor would be passing from the left high-pressure side to the right low-vapor pressure side. On the other hand, the hygroscopic properties of the material would tend to keep the moisture content on the right side at 13.5 percent and on the left side of 9 percent moisture. The moisture would naturally seek its level within the wall and would flow from left to right across the wall, or from low-vapor pressure side to the high-vapor pressure side of the wall.

The hygroscopic properties of the material and the vapor conditions of the two sides of the wall may be such that the transfer caused by the hygroscopic properties of the material is caused by vapor pressure difference. The two processes may be going on at the same time, and the final results for material can be entirely different depending upon test conditions. Thus, under one set of test conditions high vapor permeability may be recorded and under another set low vapor permeability may be recorded. This principle must be taken into consideration for the rating of hygroscopic materials.

The single word, *perm*, as suggested by the authors to be used in lieu of a complete description of conditions for reporting tests would be convenient. Were such a term decided upon, however, there should be complete agreement as to all of the units that were to be included in the expression. In other words, it might be necessary to coin a second term if some other elements or conditions were to be included.

The transfer of vapor through materials is very much more complicated than is the transfer of heat and in any rating procedure the various elements must be taken into consideration.

D. D. WILE, Los Angeles, Calif. (WRITTEN): I would like to compliment the authors for a fine presentation of a thorough development project. The simple method of determining change of weight on the inside of the sealed spaces is an excellent contribution to experimental techniques especially in view of the uniformity of results.

It would be interesting to learn if there has been any difficulty maintaining a seal between the test specimen and the ring in which it is held when working with materials which might shrink considerably during the period of the test.

Another question concerns the electric-hygrometer sensing elements used in the Model 4 Duplex Cell. Have the authors made any checks of the accuracy of these

cells under the conditions of use? For instance, has a comparison been made between these electric hygrometers and the dew-point equipment used in the Model 3 Cells?

LESTER T. AVERY, Cleveland, Ohio: Can this machine or device be used for determining the water holding content of various building wall structures?

It seems to me that the authors have a device which could be used for determining such water holding capacity. I know of nothing else which could be used for this purpose.

A. E. STACEY, JR., Syracuse, N. Y.: What is the effect of velocity of vapors over test specimens?

H. E. ZIEL, Detroit, Mich.: Would this instrumentation permit measuring of the water vapor flow through slots from a low dew-point area to a higher dew-point area?

It would be extremely helpful in the design of air conditioning systems to have some definite knowledge of the rate of water vapor flow in still and also in moving air.

AUTHORS' CLOSURE: The report requested by Dr. Carson on the project for the Office of Technical Services was published as Bulletin No. 61 of The Pennsylvania State College.[1] Fig. B from Bulletin No. 61 shows the wide variation in the permeability of wood.

Another phase of the project was concerned with the condition described by Professor Rowley. The movement of sorbed water down a gradient of moisture content without benefit of vapor pressure difference (described as transorption) was found

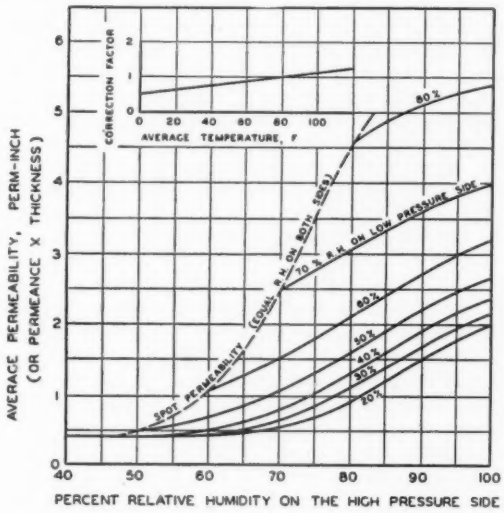

FIG. B. PERMEABILITY OF WOOD (SUGAR PINE)

[1] Water Vapor Transfer Through Building Materials, by F. A. Joy, E. R. Queer, and R. E. Schreiner (Pennsylvania State College, *Engineering Experiment Station Bulletin No.* 61, 1949). (*Editor's Note:* Bulletin is out of print but some of the material is included in *Engineering Experiment Station Technical Paper No.* 89, Basic Concepts of Water Vapor Migration and Their Application to Frame Walls, by F. A. Joy).

to be slight under the conditions of the test. Later work by R. C. Calvert[2] demonstrates clearly that transorption is nonexistent in a low range of sorbed moisture but, in the case of free water, there is good reason to expect capillary movement.

The unit *perm* is intended as a convenient substitute for the long form, grains per (sq ft) (hr) (in. Hg). This is the full meaning of *perm* and the term does not imply any description of (test) conditions as suggested by Professor Rowley. Since the permeance of some specimens depends on exposure, a statement of test conditions should always accompany a report of permeance.

Mr. Wile notes two very practical difficulties. The shrinkage or warping of a heavy specimen sometimes requires a period of exposure to shape it before test data can be taken. The electric hygrometer elements are recalibrated as required. They are more convenient than the dew cup but the latter is of course a basic measurement.

Mr. Avery suggests another good use for the cell. The safe water holding capacity of materials is often an important factor in a condensation control problem.

The effect of air velocity, as suggested by Mr. Stacey, is considerable in a test of a high permeance specimen, but it is slight for barriers whose permeance is less than one perm. The vapor resistance of air is relatively small.

Mr. Ziel's question about slots and cracks is very practical. The cell can be used for some of the simplest problems but full scale tests are better. Vapor movement through cracks is generally not a matter of diffusion. Transportation of vapor by a moving air mass is likely to be the important mechanism in such cases.

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[2] An Investigation of Transorption, by R. C. Calvert (A thesis presented for the degree of M.S. in Mechanical Engineering, The Pennsylvania State College).

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No. 1372

HUMAN TOLERANCE LIMITS FOR EXTREME HEAT

By W. V. Blockley* and Craig L. Taylor,** Los Angeles, Calif.

The advent of aircraft with flight speeds in the sonic range presents the potential hazard of heat to pilots and occupants. In the event of failure of the air conditioning system, air entering the cabin would be heated in proportion to the square of the aircraft's speed. The orders of magnitude to be expected may be indicated by the following example: at low altitude on a 100 F summer day, air temperature in the cabin at a speed of 800 mph would rapidly reach 215 F.

Tolerance for extreme temperature is also a practical problem of industrial physiology and hygiene. Under special, though usually emergency, conditions workmen may be exposed to dry or humid heat. One of the authors has received a number of communications from individuals recounting such experiences. Though unverified, these accounts are adequately factual and not inconsistent with the results generally obtained in experimental situations. Thus, a kiln technician reports many exposures for two or three minutes to 250 F, and on several occasions to 500 F. A plastics engineer regularly spends 10 min out of each 30 in an oven at 200 F. A mining engineer tells of rescue parties exploring pockets in a burning mine where temperatures ranging up to 240 F were encountered. These experiences outline the need for a knowledge of the human limitations, practical measures of personal protection, and for the development of efficient protective equipment.

In the literature of investigations into man's ability to withstand extreme thermal environments, accounts appear as early as 1775, when Charles Blagden and George Fordyce[1] reported on their experiments in heated rooms at temperatures up to 260 F. In modern times, such studies have been mainly concerned with conditions involving high humidities, but only moderately high dry bulb temperatures. Outstanding in the contributions to this field are the researches begun in the early 1920's at the A.S.H.V.E. Laboratory in Pittsburgh, and one important aim of this paper will be to compare the results obtained in the present study with those of the A.S.H.V.E. group. Other workers have studied reactions of working men to heat and high humidity, notably Dreosti[2], Eichna et al[3] and Robinson[4].

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Minneapolis, Minn., June 1949.

[1] Exponent numerals refer to References.

EXPERIMENTAL CONDITIONS AND PROCEDURE

The heat exposures discussed in this paper were carried out early in 1948 as part of a general program of research into man's ability to tolerate extreme heat.†

The experimental heat chamber is an insulated cylinder, 70 in. × 46 in. in diameter, with a total volume of 70 cu ft. Ventilation and heating are accomplished by means of a blower, flow metering system, and a 9 kw electric heat exchanger. The flow through the chamber is distributed over its horizontal cross-section by a floor baffle plate system. Its equivalent linear velocity, directed vertically from seat-to-head of the subject, averaged 41 fpm as measured with a thermal anemometer. The ventilation volume rate was maintained, as a matter of routine, at 70 cfm, exhaust air being discarded.

Air temperature was controlled by a thermocouple-potentiometer system with sensing element in the chamber. Wall temperature remained within 10 F deg of air temperature under all conditions. Humidity was not controlled, and the vapor pressure of the chamber air was a function of outdoor humidity and moisture added by the subject, ranging generally between 0.5 and 1.0 in. Hg and averaging about 0.8 in. Hg.

Two student volunteers were subjected to environments ranging in dry bulb temperature from 140 F to 240 F inclusive. In all these experiments, clothing was standardized, and consisted of a close-fitting wool and cotton one-piece union suit, 0.080 in. thick, with loose-fitting felt duffel socks on the feet.

Before each heat exposure, subjects spent at least 30 min resting in a comfort environment of approximately 80 F to establish baseline values for the physiological measures which were as follows:

1. Rectal temperature (t_r) obtained by a thermocouple inserted to 7 in.

2. Skin temperature, measured at nine locations and weighted according to the DuBois surface area factors to yield the mean skin temperature (t_s).

3. Exhaled air temperature, (t_e) measured at the mouth portal during channelled breathing (mouth exhalation).

4. Heart rate (H.R.) taken by means of 30 sec pulse counts at the wrist by the subject or counted from electrocardiographic records.

5. Respiration volume, oxygen consumption and respiration rate, taken from records of a closed circuit metabolism apparatus.

6. Blood pressure determined by standard auscultatory technique.

During the exposure period, determinations of the foregoing measures were made as frequently as possible—approximately every two to three minutes. Above 140 F, the subject remained in the test environment until the tolerance end point was reached as defined by the following:

1. Subject's report of intolerable discomfort. Under conditions of intelligent and manful cooperation from our subjects (which was at all times highly satisfactory), the authors believe that a practical limit was achieved. Certainly, there was no evidence of terminating on whim or personal inconvenience. Sensations of heat or burning were never sole criteria for tolerance.

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† Work performed under contract with the U. S. Air Force, Air Materiel Command, Aero-Medical Laboratory, Wright-Patterson Air Force Base, Dayton, Ohio.

2. Appearance of symptoms of hyperpyrexia and heat exhaustion, including faintness, nausea, tingling sensations in extremities, mental confusion, compulsive restlessness, dyspnea or symptoms referred to the chest, when clearly developed.

3. Heart rate of 150 beats per minute. This was a physiological sign influencing the decision, but was not used as an absolute criterion; that is, in the presence of other signs, experiments might be terminated at heart rates less than 150, but, in some cases, 150 was exceeded when the subject was otherwise in good shape.

TABLE 1. ENVIRONMENTAL CONDITIONS AND TOLERANCE TIME

MEAN AIR TEMPERATURE (DEG F)	NUMBER OF EXPOSURES	RANGE OF INLET VAPOR PRESSURES (IN. HG)	RANGE OF TOLERANCE TIME (MINUTES)	AVERAGE TOLERANCE (MINUTES)
140-142	2	0.075-0.36	63-64[a]	63.5[a]
154	1	0.12	79[a]	79[a]
179-181	4	0.14-0.24	44-53	48.5
199-203	4	0.15-0.32	31-35	33
218-224	6	0.17-0.38	25-30	28
237-242	4	0.14-0.51	20-26	23.5

[a] Terminated before tolerance was reached.

TABLE 2. PHYSIOLOGICAL RESULTS OBTAINED IN A TYPICAL EXPERIMENT

EXPERIMENTAL TIME (MIN)	AVERAGE AIR TEMP. (DEG F)	RECTAL TEMP. (DEG F)	EXPIRED AIR TEMP. (DEG F)	MEAN SKIN TEMP. (DEG F)	HEART RATE	BLOOD PRESSURE (MM HG)
Pre-exposure (start = 0 time)						
30	80	97.8	95.7	95.0	80	128/80
Exposure (start = 0 time)						
3	228			100.5	104	
6	238	98.2	98.4	101.5		
10	245	98.6	100.8	102.1	114	
15	245	99.3		103.3		
18	242	99.9	102.8	104.1	150	
24	242	100.8	105.6	105.0	150	
25		101.1				
26 Tolerance Time.						
Recovery (time continued)						
29	75				132	180/90/60
41	75	99.3		95.3	88	100/50
44	75	100.4	98.0	96.8	92	118/80
62	75		95.7	93.2		122/88

The elapsed exposure time to this end point is referred to as *tolerance time*. It is significant that, despite the inexact nature of this definition of tolerance, the tolerance times observed in different subjects at similar environmental conditions were invariably within 10 or 15 percent of each other.

The listed physiological responses were followed during the recovery after termination of the exposure.

RESULTS

In Table 1 are summarized the environmental conditions and tolerance times for the 21 experiments of this series.

In Table 2 are shown physiological results obtained in a typical experiment, No. B-13, mean average air temperature 240 F; the detailed responses with time are graphed in Fig. 1.

The authors plan to publish separately a paper†† dealing in detail with the physiological aspects of heat exposure, but the following is a summary of findings up to the present time.

1. Skin temperature rises continuously during exposure, the rate of rise being successively greater with increasing ambient temperature levels. The maximum mean skin temperature observed was 107 F, reached in an exposure at 240 F.

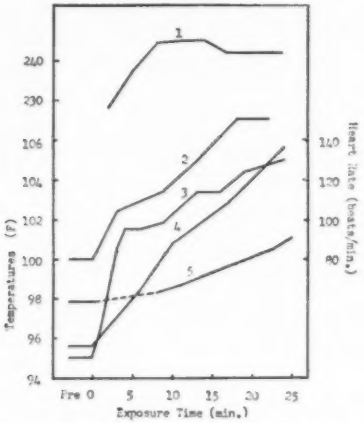

FIG. 1. EXPERIMENT B-13: CURVE 1, AIR TEMPERATURE; 2, HEART RATE; 3, MEAN SKIN TEMPERATURE; 4, EXPIRED AIR TEMPERATURE; 5, RECTAL TEMPERATURE

2. Heart rate increases in a similar fashion to skin temperature, usually reaching values between 140 and 160 beats per minute.

3. Blood pressure changes show greater individual differences than do other physiological factors. Systolic pressure rose during exposure in all cases, while diastolic pressure declined continuously in one subject, but remained relatively stable or slightly elevated in another.

4. Electrocardiograms taken during and after heat exposure show minor variations from the normal resting records of the individual, but no distinct indications of heart damage have been seen.

5. Rectal temperature changes are relatively slight and slow in onset as compared with other physiological reactions, and are an unreliable indication of the extent of heat stress on the body under these conditions of short exposure.

6. Exhaled air temperature responds rapidly to the heat load, and appears to be a useful indication of thermal stress. Measurement of temperatures at several loca-

†† The work on which this paper is based is more fully described in an Air Force Technical Report entitled *Studies of Human Tolerance to Extreme Heat*—First Summary Report.

tions in the respiratory passages indicates that the mucous membranes exert a strong regenerator influence, cooling the inhaled air by as much as 100 F in a few inches of travel, and warming the exhaled air, though to a much lesser degree. However, as far as can be determined, because of the evaporative loss the total respiratory system continues to serve as an avenue of heat loss even under conditions close to the extremes of these studies.

7. Respiratory ventilation increases markedly under heat stress, often reaching values two or three times the resting levels, though respiration rate is not significantly affected. Oxygen consumption apparently follows a course roughly parallel to that of ventilation.

TABLE 3. INCREASE OF SURFACE AND RECTAL TEMPERATURE WITH EFFECTIVE TEMPERATURE

EFFECTIVE TEMPERATURE	Δt_r (DEG F/HR)	t_r	Δt_s (DEG F/HR)	t_s	ΔHR (BEATS/MIN/HR)
90	0.4	99.0	0.4	96.4	3
95	0.9	99.5	1.2	97.2	11
100	2.1	100.7	3.3	99.3	42
105	4.0	102.6	8.8	104.8	98

Δt_r (increase in rectal temperature) and ΔHR (increase in heart rate) values were read from Fig. 5 and Fig. 6 (Houghten and Yaglou[5]).

Δt_s (increase in skin temperature) was calculated from Fig. 3 (McConnell and Yaglou[4]).

$t_r = 98.6 + \Delta t_r$; $t_s = 96.0 + \Delta t_s$; one hour exposure assumed.

8. The rate of sweat loss is highly variable, but tends to increase with successively higher levels of ambient temperature. The rate of actual evaporation increases with air temperature at a considerably slower rate than does the total weight loss. Absolute values of the weight loss observed range from 0.9 to 2.5 lb depending on the characteristics of the individual, the duration of exposure, and the ambient temperature level.

9. Subjective symptoms of thermal strain are numerous, but only a relatively few were observed consistently in each experiment. Some sensations, while definite, are difficult to describe in precise terms. One common feature of all exposures carried to tolerance is a feeling of air hunger, associated with deep and irregular respiration. This symptom may be accompanied by restlessness and nervous irritability, while waves of dizziness may intervene at terminal stages.

Items 1, 5 and 6 deserve emphasis, since they illustrate the main difference in effect between the short-term, high dry bulb environments studied by the authors and the high humidity, moderate dry bulb environments which were used in the majority of the Pittsburgh A.S.H.V.E. studies. Table 3 illustrates the tendency of surface temperature to outstrip rectal temperature as the severity of the heat load is increased.

In the authors' experiments, the maximum rectal temperature recorded was 101.7 F. In many cases, the peak temperature was reached after exposure, during the recovery period, and in several exposures it did not exceed the normal range.

INDEX OF PHYSIOLOGICAL STRAIN

For the purpose of clarifying the relationship between physiological responses and air temperature, the collected data were analyzed at a particular exposure

time. Exposure time 20 min was selected as the comparison point, in order to avoid early adjustment effects and yet include all experiments. Values for the various physiological measures at 20 min were read from the response-time curves, while in the case of air temperature the value for the whole experiment was used.

The correlation coefficients are shown in the following tabulation:

(1) Heart rate with air temperature	0.91
(2) Mean skin temperature with air temperature	0.96
(3) Rectal temperature with air temperature	0.64
(4) Exhaled air temperature with air temperature	0.89
(1) + (2)[a] with air temperature	0.96
(1) + (2) + (3) with air temperature	0.92
(1) + (2) + (4) with air temperature	0.98

[a]For the purpose of the combinations, Z scores were computed according to the formula, $Z = (x - M_x)/S_x$, where x = a variate, M_x = the mean and S_x = standard deviation of the distribution of the variates. The Z score corresponds to the *relative deviate scale*, k, as defined and derived by Treloar, *Elements of Statistical Reasoning*, 1939; it allows the expression of a series of quantities as pure numbers, so that the problem of dissimilar units is eliminated. All other statistical methods and symbols used in this paper are fully explained in the latter work.

The results show that the combination of skin temperature and heart rate, equally weighted, yields a coefficient of 0.96. Adding t_e slightly improves the relationship, but this combination was discarded because of the fewer cases containing determinations of t_e. Rectal temperature reduced the relationship, as might be expected from the preceding results.

If it is assumed that the conclusions from the analysis of 20-min data hold approximately for any time up to 35 min, a simple index of physiological strain can be derived, namely

$$\text{Index} = \tfrac{1}{2}\,(Z_{t_s} + Z_{HR})$$

To this end, means and standard deviations for the total distributions ($N = 102$) of skin temperature and heart rate were calculated, as follows:

HR: mean = 109.4; standard deviation = 18.26
t_s mean = 101.6; standard deviation = 2.53

Substituting and simplifying:

$$\text{Index} = 0.0274\,(\text{HR}) + 0.198\,(t_s) - 23.10.$$

SENSIBLE HEAT TRANSFER

In the experiments reported here, it was not possible to incorporate routine measurements of evaporative heat loss and metabolism, and consequently complete heat balances cannot be reported. Radiant and convective heat transfers from chamber walls and air to clothing surface have been calculated by the equation,

$$q_r + q_c{}^* = h_r\,(t_w - t_g) + h_c\,(t_a - t_g)$$

*It will be noted that for clarity in presenting a portion of the human heat balance the equations have been rearranged to give a positive sign to $q_r + q_c$. In conventional treatments of the total heat balance, this quantity would bear a negative sign when the environment is hotter than the body surface (see HEATING, VENTILATING, AIR CONDITIONING GUIDE 1949, p. 216).

where

q_r, q_c = radiant and convective heat transfers Btu per (square foot) (hour).
t_g, t_a, t_w = garment surface, air and wall temperatures (Fahrenheit degree).

Since, in a steady state, the same quantity of heat must be conducted through the clothing, this relation is expressed:

$$q_r + q_c = h_g (t_g - t_s)$$

where

t_s = skin surface temperature (Fahrenheit degree).

Solving the two equations for t_g, setting them equal, and solving for $q_r + q_c$, gives the expression for overall heat transfer from environment to body:

$$q_r + q_c = \frac{h_g (h_r t_w + h_c t_a - h_r t_s - h_c t_s)}{h_c + h_r + h_g}$$

where

h_g = unit conductance of garment, Btu per (square foot) (hour) (Fahrenheit degree temperature difference between surfaces of clothing).
h_r = unit heat transfer by radiation, Btu per (square foot) (hour) (Fahrenheit degree temperature difference between wall and garment).
h_c = unit conductance for thermal convection, Btu per (square foot) (hour) (Fahrenheit degree temperature difference between air and garment surface).

Coefficients of heat transfer h_g, h_r, h_c for the test conditions reported have been adapted as follows:

$$h_r = \frac{\sigma F_{ea} (A_r/A_b) (T_g^4 - T_w^4)}{t_g - t_w}$$

Btu per (square foot) (hour) (Fahrenheit degree per foot).

where

F_{ea} = 0.57 (this shape-emissivity modulus is based upon the emissivities of garment and walls).
σ = Stefan-Boltzmann radiation constant.
A_r/A_b = 0.75 (the fraction of body area concerned in radiation as taken from Winslow et al).
h_c = 0.020 $V^{1/2}$ Btu per (square foot) (hour) (Fahrenheit degree).

where

V = air velocity feet per hour. This equation is also taken from Winslow et al[7], and modified to the temperatures (200 F) of these experiments.

$$h_g = k \frac{1}{L}$$

where

k = thermal conductivity of garment Btu per (square foot) (hour) (Fahrenheit degree per foot).
L = garment thickness, feet. According to Schiefer et al[8] k for the 50:50 cotton-wool underwear used in these experiments is 0.028, and L, from the authors' measurements at 0.1 lb per square inch is 6.83×10^{-3} ft.

Values of $q_r + q_c$ then represent an external heat load expression, with which the physiological effects and tolerances may be correlated.

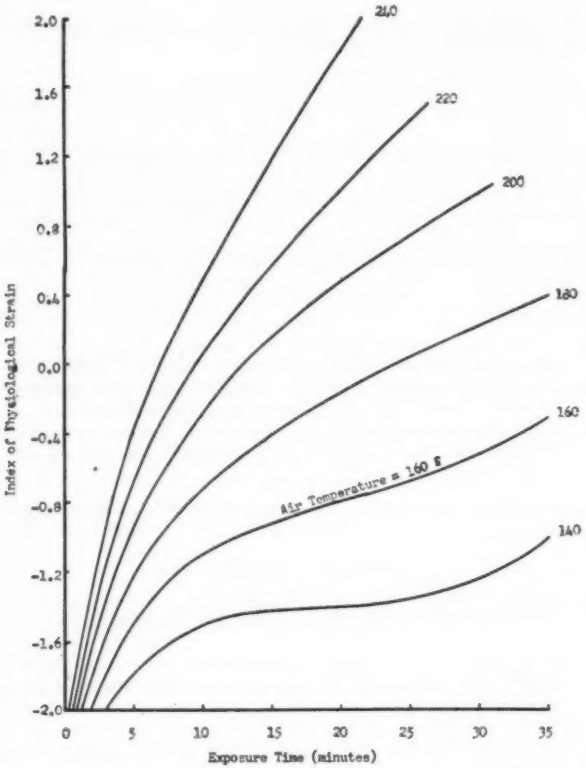

FIG. 2. RELATIONSHIPS BETWEEN AVERAGE INDEX, AIR TEMPERATURE, AND TIME OF EXPOSURE

Data are based upon two subjects exposed a total of 18 times, approximately two to three times to each air temperature

RELATIONS BETWEEN HEAT LOAD CRITERIA AND INDEX OF PHYSIOLOGICAL STRAIN

The data from 18 experiments were separately analyzed at 5, 10, 15, ... 35 min of exposure time as a means of determining the relations between heat load criteria and index of physiological strain. Coefficients of correlation were obtained for each 5 min period, (*a*) between chamber air temperature and Index,

and (*b*) between $q_r + q_c$ and Index. It was found that these coefficients agreed closely throughout. For example, at 20 min, correlation (*a*) was 0.96, correlation (*b*) was 0.95. From this it was concluded that the present data could most conveniently be treated in terms of the air temperature criterion.

RELATIONS BETWEEN EXPOSURE TIME, AIR TEMPERATURE, AND STRAIN INDEX

Since, in the temperature range concerned, the practical problem is one of time-tolerance, it is necessary (*a*) to examine the changes in physiological state with time, and (*b*) to determine the relationships between the maximum tolerable index of strain, exposure time, and air temperature.

The first of these comparisons is displayed in Fig. 2, a plot of Index versus time for the various air temperature groups. Since the exposure temperatures were originally planned for 20 F intervals from 140 to 240 F (except 160), and the mean air temperatures actually obtained seldom varied more than ± 3 deg from the designated temperature, it is possible to construct a simple family of curves. The points along each 5 min time line were computed from the least square linear fits made between air temperature and Index for each 5 min block of data. Because of the data at hand, this time function is presented graphically. With more experiments, mathematical analysis of time trends will be justified. It is clear that increasing emphasis must be given to time rates for each mode of adaptation in evaluating tolerance for short intense exposures.

TOLERANCE CONTOURS

With a view toward displaying the salient features, Fig. 3 has been constructed by reading points from Fig. 2, presenting Index as a function of ambient air temperature and exposure duration. Plotted upon this figure are the actual tolerance times of the experiments. One immediately sees that these tolerance times do not correspond to any constant Index level, but appear to increase in Index with air temperature. This means that factors additional to heart rate and skin temperature have to do with the tolerance time. It will be recalled from the preceding discussion that numerous and diverse symptoms are associated with tolerance. These are not necessarily highly correlated with Index.

For comparison, the results of two other investigations featuring suitably similar conditions of exposure have been included in Fig. 3. Tolerance times averaged from still air, 200 fpm and 400 fpm air velocity exposures at various effective temperatures have been taken from McConnell, Houghten and Yaglou[9], Table 3. These differences in air movement apparently had no effect upon tolerance time, according to the authors, and advantage may be taken therefore of the larger number of cases in their combined[10] results. The effective temperature chart was used to extrapolate the effective temperatures to the equivalent temperature at a vapor pressure of 0.8 in. Hg which is the average of our experiments. This temperature, together with the tolerance time, could then be plotted on Fig. 3.

Results from Wright Field exposures[10] are also plotted. These give a point at 60 min, which represents tolerance for 7 experiments, and an average toler-

ance time of 39 min obtained in 5 other experiments. These tolerance times were also entered at an ambient temperature (at 0.8 in. Hg vapor pressure) equivalent in physiological effect to those of the actual exposure. An equation of equivalence, given in the reference cited, was used to make the necessary conversions.

The plots of Fig. 3 show that there is fair general agreement between the three sets of data, although it should be noted that the 39 min tolerance point

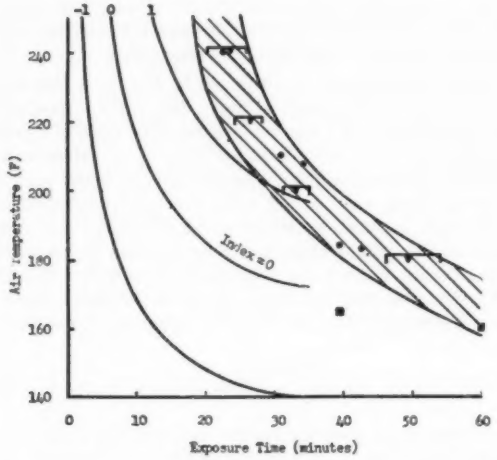

FIG. 3. CONTOURS OF INDEX AND TOLERANCE TIME. BRACKETS AND ARROWS INDICATE RANGE AND MEANS OF UCLA TOLERANCE TIMES; SOLID CIRCLES, McCONNELL ET AL[9] (SEE TEXT FOR EXPLANATION); SOLID SQUARES, TAYLOR ET AL[10]

from the Wright Field data is apparently at variance. There is no immediate explanation for this discrepancy. The cross-shaded area represents a zone, within which individuals will reach tolerance at some time depending upon motivation, state of fitness, and numerous other factors which are as yet obscure. No attempt has been made to formulate the shape of the curve, nor has a safe tolerance line been established, beyond that implied in the inner limit of the shaded area. Such a tolerance limit will be offered when more experimental results have been accumulated.

It should be emphasized that the tolerances strictly apply only to the conditions of the experiments, particularly with respect to the limited humidities. Advantage, however, may be taken of the general agreement with A.S.H.V.E. and Wright Field results to extend the tolerance lines to other temperature and humidity combinations, with the aid of the criteria of equal effect explained in the original papers.

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DISCUSSION

E. F. DUBOIS†, M.D., New York, N.Y. (WRITTEN): This is a valuable study of short exposures. The experimental method employed is difficult and not without danger, and it will be hard to get more volunteers as subjects. If more men were studied, there would undoubtedly be wide variations and probably instances of very low tolerance times. With longer exposures, the rectal temperature would certainly take on more importance. When the skin and inspired air are warmer than the interior of the body, there is obviously no method of dissipating the heat produced within the body.

It would be interesting to follow the efficiency of the subjects during these exposures and determine the length of time that a man would be in shape to pilot a plane. The impairment of efficiency in dry and humid heat is a limiting factor in airplanes and submarines.

A. P. GAGGE*, Dayton, Ohio. (WRITTEN): The authors are to be congratulated on the work reported here, for their courage in undertaking it, for their ingenuity in the design of their experiments and their necessarily complex equipment, and for the exhaustive analysis and integration to which they have subjected their data. Especially noteworthy is the authors' success in correlating significant physical and physiological variables.

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* Lt. Colonel, MSC (USAF), Chief, Aero Medical Operations, Aero Medical Laboratory, Engineering Division, Wright-Patterson Air Force Base.

It would appear that these authors have been handicapped by the lack of physiological and psychological definitions of tolerance, as are most workers in the fields of stress physiology and psychology. It is remarkable that good correlations have been found which are based on an admittedly arbitrary and incomplete definition of tolerance. From the existence of these correlations, and others like them, it is possible to hope that more complete definitions of tolerance, which are based on appraisals of both physiological and psychological factors, may be forthcoming in the not too distant future. But the realization of this hope will require the concerted efforts of physiologists and psychologists. It is likely that the most troublesome impediment to this goal will be the lack of quantitative measures of motivation.

There is much need for the type of study carried out by these authors, and it is hoped that their contributions to basic and applied science will continue. It will be interesting to follow the extension of this research, and its application to industrial and military situations, to see what relation the tolerance, as defined here, bears to what might be termed operational tolerance. Now that their major problems of equipment and technique are behind them, the authors may wish to devote time and thought to the possibilities of making their results generally applicable to industrial and military problems, and toward finding the factors which cause the deviations between their index curves and the tolerance curve.

L. P. HERRINGTON‡, New Haven, Conn. (WRITTEN): This very useful study fills an important gap in the data of industrial medicine and hygiene. Many industrial processes require the heating engineer to provide, in local spaces, temperature conditions which are known to be completely beyond the steady state adjustment possibilities of human temperature regulation. Although industrial medical consultants have recognized that, as a class, individuals with a history of circulatory impairment should not be permitted to take even short exposures at temperatures beyond 100 F, this has not solved the more general problem of a sensible exposure rule for average personnel.

Under such circumstances, casual worker experience has established certain rough conventions with respect to exposure time which work very well as local rules, but are of little use to the designer or consultant requiring systematic information.

The straightforward approach to this problem by Blockley and Taylor has resulted in our first systematic data on short term human heat tolerance. The investigation has obviously been conducted with a high degree of scientific competence, and very adequate technical equipment.

The decidedly unpleasant and strenuous nature of the human experience required to record these data, no doubt, is in large part responsible for the fact that it is not possible to match this report with earlier data of equal value. Under these circumstances, perhaps it is ungenerous to suggest further work which would be of industrial interest. However, we may hope that these experimenters may give us future results in which water vapor content is varied over a wider range. The thermal shock of high vapor content at elevated temperatures must be considerable, and is subjectively rather different from that experienced in low humidity. However, the present report deals most constructively with what is by far the most important issue; namely, tolerance time for limited exposure to hot and relatively dry atmospheres.

Management, medical, or engineering personnel who may find a use for these results should be warned that there is a vast difference between tolerance time as here determined and permissible recurrent exposure time. All such exposures are dyshygienic in character, and are to be discouraged in principle. What we have in these data are upper limits which can be reached by men in good condition, and it is extremely valuable to have this reference point. In practice, the limitation of

‡ Director of Research, John B. Pierce Foundation, Laboratory of Hygiene.

occasional exposure to 25 percent of these limits, and repeated exposure to 10 percent, might be suggested. While this is arbitrary as to percentages suggested, it is in agreement with the sound hygienic principle of working near the lower rather than the upper limit of safety.

C. M. HUMPHREYS, Cleveland, Ohio. (WRITTEN): The authors are to be congratulated for presenting a paper which adds considerably to our knowledge on human tolerance of hot environments.

Through the medium of the effective temperature concepts, the authors have compared their results with those obtained a number of years ago by W. J. McConnell, F. C. Houghten, and C. P. Yaglou at the A.S.H.V.E. Laboratory, and excellent correlation was obtained. Fig. 3 of the paper shows this correlation between air temperatures of approximately 180 to 240 F, or 106 to 114 F E.T. (effective temperature). Additional data which were given in the A.S.H.V.E. Laboratory paper would permit the extension of the smooth curve of Fig. 3 to a dry bulb temperature of 143 F, (98.2 E.T.) and an exposure time of almost three hours. This is just one more unexpected verification of the upper region of the effective temperature chart.

An unsuccessful attempt has been made to correlate the results presented in this paper with those obtained in the work which was done for the U. S. Navy and presented to the Society in 1946. However, the lack of correlation is not surprising. The Navy investigation was made to determine the most severe conditions under which the average man could remain alert and active throughout a 4-hr watch. Therefore, the subjects were active, not seated at rest, and the limiting values for pulse rate and temperature were set somewhat lower.

It was found in the work done for the Navy that the degree of acclimatization acquired had a marked effect upon the ability of the subjects to endure the conditions imposed. I should like to ask the authors if any attempt was made to acclimatize their subject to hot environment prior to the series of tests reported in the paper. If not, then it would seem reasonable to believe that the acclimatization acquired during the early part of the testing program may have materially increased the endurance times determined in the latter part of the program.

C.-E. A. WINSLOW, New Haven, Conn. (WRITTEN): The authors have added to our knowledge of the influence of temperature on human life by a well-planned study of the ultimate tolerance limits of lightly dressed subjects at rest exposed to extremely high dry-bulb temperatures—far beyond the ranges which have ordinarily been observed.

Of particular interest are the very high correlations of mean skin temperature and heart rate with air temperature (over 0.9 in each case). It is significant, too, to note in Fig. 2 the fact that, at 140 F and even to less extent, and at 160 F, the index of physiological strain plotted against time levels off for the first half hour, no doubt as a result of increased evaporative efficiency.

It is a striking observation that, after exposure to about 240 F air temperature for 24 min, the rectal temperature should have reached 100.8 deg, the mean skin temperature, 105.0 deg, and the heart rate, 150 beats. High humidity, of course, would even further decrease the tolerance times observed.

AUTHORS' CLOSURE: In reply to the suggestions of several of the discussers, we hasten to state that the omission of studies of the effect of humidity in the first group of experiments reported here was due merely to the lack of humidity control equipment. This has been remedied, and a systematic exploration of humidity effects over the practical range of climatic vapor pressures, 0.4 to 1.4 in. Hg., is in progress during the present year, 1948-49. Results show the familiar thermo-mimetic effect of humidity which can be expressed either in equivalent temperature lines on a

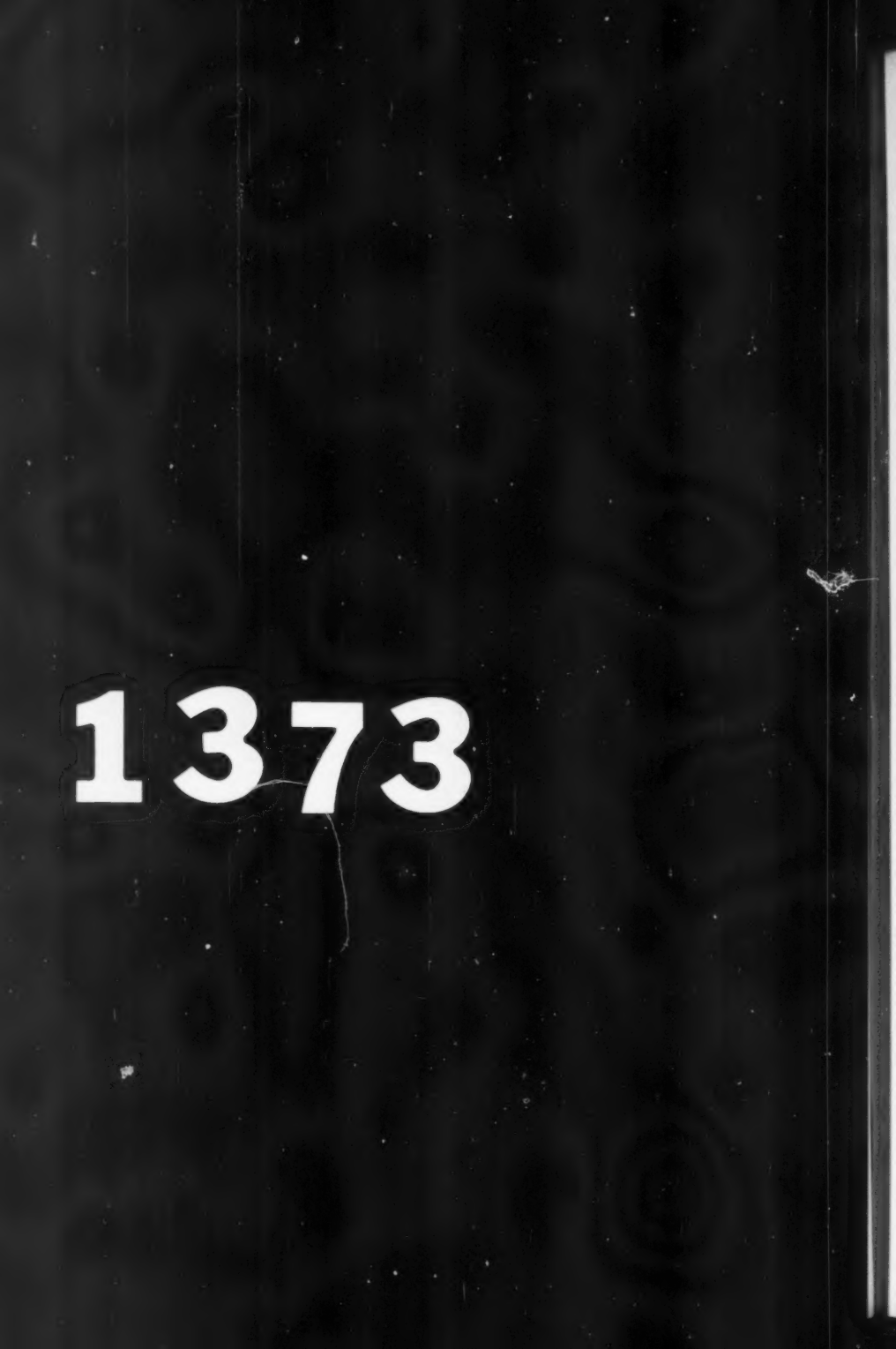
psychrometric chart or in rational quantitative expressions of sensible and latent heat transfer.

Dr. DuBois and Colonel Gagge indicate the desirability of making observations on functional properties during exposure to heat. It is our plan to attempt first the formulation of biophysical and physiological tolerances, then to study sensory, motor, and intellective functions as affected by the thermal environment. If it is found that significant decrements occur in advance of the physiological signs and criteria, a redefinition of tolerance may be indicated for specific operational situations. In the program for 1949-50, investigations of these questions are planned.

The comment of Dr. Herrington on short term tolerance brings out a fact worthy of further remark: namely, that the investigation of high temperature tolerance is basically one of the transient state of human thermal adaptation. Previous investigations, such as those of the Russell Sage and the Pierce Laboratories, have been chiefly devoted to the steady state, and understandably so, because the practical ends in view have concerned adaptations to thermal exposures of hours in duration. Here, however, time limits are critical, and rates of change in the various modes of adjustment strongly affect individual tolerance times. This situation necessitates special treatment of instrumentation, experimental procedure, and analysis of data to elucidate the transient state. We are giving special attention to the time course of events in treating our present results.

The index cited in the paper simply refers to a combination of heart rate and skin temperature which has shown validity in plotting the individual's response to heat. This index, however, is not to be extrapolated beyond the present data. It is in no sense to be regarded as a final criterion of the magnitude of internal thermal strain, nor as a perfected tool for rating the tolerability of environments. Much more work must be done before a trustworthy guide of this sort can be offered.

We prefer to consider our subjects, who have had from five to ten exposures to the heat, as *habituated*. To what extent this condition approaches the *acclimatized* state referred to by Mr. Humphreys, we do not know. Experiments and observations on acclimatization commonly involve weeks and months of exposure, and, usually, the added stress of exercise. We have data on several subjects at present and will gather further observations on the process of habituation to extreme temperature.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No. 1373

HEAT GAINS ARE NOT COOLING LOADS†

By C. O. MACKEY* AND N. R. GAY**, ITHACA, N. Y.

COOLING loads are commonly calculated as if the energy transferred from lights, appliances, occupants and warm surfaces and through glass found its way into the *air* immediately and without reduction. Part of the energy emitted from such sources is in the form of radiation and part is in the form of heat transferred directly to the air by convection. If remote surface cooling of the conditioned air is used (cooling coils or spray chambers), only the convected energy is an immediate and direct cooling load.

Radiation emitted from the sources mentioned is incident upon various surfaces, such as floors, walls, ceilings, and furnishings within the enclosure. The direct absorption of this radiation by carbon dioxide and water vapor and the heat storage capacity of the air are usually negligible in relation to the other effects under consideration.

Radiation incident upon surfaces which do not transmit radiation directly is largely absorbed, but partly reflected, at these surfaces. The energy absorbed may then be (1) stored in the materials with accompanying rise of temperature, (2) conducted through the materials in the direction of decreasing temperature, (3) transferred by radiation to other surfaces at lower surface temperatures, or (4) transferred by convection to the room air when the temperature of the surface exceeds that of the air.

If there is no direct cooling such as is obtained when cold panels in the conditioned space absorb radiation emitted by the primary source, only the heat transferred by convection from the secondary absorbing surfaces to the room air is an immediate cooling load in that space. It is evident that a unique energy balance must exist before the cooling load within the enclosure is equal to the rate of energy emission from primary sources therein.

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† This study has been made at the suggestion of the Technical Advisory Committee on Cooling Load—W. E. Zieber, *chairman;* C. M. Ashley, R. D. Blum, W. F. Friend, R. H. Heilman, H. W. Heisterkamp, R. C. Jordan, C. F. Kayan, J. N. Livermore, R. H. Lock, C. O. Mackey, J. P. Stewart.

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STORAGE EFFECTS

Because large quantities of energy are stored in furnishings and building materials, this unique energy balance seldom exists, and the cooling load at a given time is seldom the same as the instantaneous total rate of energy emission from the various sources. Therefore, *heat gains are not cooling loads.*

The common building with its furnishings approaches a constant-temperature energy reservoir. Large quantities of energy are stored with small temperature rise because of the high product of weight and specific heat of the building materials and furnishings.

For example, assume a total rate of energy emission from all sources within an enclosure of 60 Btu per hour for each square foot of floor space. Assume that one-half of this energy is emitted as radiation and one-half immediately transferred to the air by convection; the furnishings and bounding materials of construction might easily weigh 150 lb per square foot of floor space and have a specific heat of 0.2 Btu per pound per Fahrenheit degree. If all the energy emitted by radiation within the space were stored, the temperature of the storing materials would rise at the average rate of only 1 deg per hour.

Available information on cooling loads consists almost entirely of the total rates of emission of energy by various sources. In short, the engineer has little information on cooling loads. He may be able to find (1) the rates of emission of energy from lights, appliances, people, and warm surfaces and (2) the rate of transmission of energy by glass. Even if he realizes that appreciable portions of this energy are in the form of radiation, he has little data from which to estimate real cooling loads. Not only is he handicapped in the determination of capacities of equipment but when the possibilities of pre-cooling and after-cooling are considered, he has little data concerning the most economic or optimum method of operating the equipment installed.

So many variables, in addition to the energy release rates at that time, influence the true cooling load at any particular time that the determination of cooling load does not lend itself to an exact analytical approach. It is not the purpose of this paper to develop any general, analytical method of solution. Instead, a few examples with simplifying assumptions will be presented and true cooling loads will be estimated and compared with instantaneous rates of emission of energy. From these examples, tentative general conclusions will be drawn, but it must be admitted at once that much more study would be necessary before these conclusions would be completely adequate and entirely sound.

SOURCES OF ENERGY EMISSION

For purposes of estimating cooling loads, sources of energy emission within an enclosure may be divided into two groups. One group might include the sources from which the rate of energy emission remains substantially constant with respect to time—*the steady source.* In this group might be placed, for first analysis, people, lighting, and appliances. The second group might include the sources from which the rate of energy emission varies with respect to time—*the unsteady source.* In this group might be placed warm interior surfaces of walls or ceilings and glass areas transmitting solar radiation. If the design day is to be repeated in cyclic fashion, these unsteady sources may generally be classified as periodic with a period of 24 hours.

Steady Sources of Energy: Among the common sources of energy that are usually assumed to be steady in estimating cooling loads are appliances, lights, and people. Some attention has been given to the relation between the rate of input of electrical energy to lights and the resulting cooling load. Measuring cooling loads in a Detroit office building, J. N. Livermore[1] concluded that:

. . . . in calculating the cooling load due to lighting, there should be a time lag allowance for heating up of building materials with a varying proportion of the energy input going directly to the atmosphere.

Mr. Livermore also found that the greatest apparent differences between true cooling loads and instantaneous rates of energy emission occurred with heavy lighting loads. The calculated peak loads were generally found to be higher than the measured loads.

In interesting studies of a pioneering nature, C. S. Leopold[2], employing simplifying concepts, has used an hydraulic analogue to determine the relation between cooling loads and instantaneous rates of energy input to lights in specific examples.

The authors have used the mathematical equivalent of the analogue, designated as the Schmidt method, in three specific examples of steady sources of energy emission.

EFFECT OF LIGHTING LOAD

Example 1: A lighting load with a steady rate of input of electrical energy of 11.6 Btu per hour (3.4 watts) for each square foot of floor area was analyzed. Following Mr. Leopold, the authors assumed the lighting to be located in each floor of a multi-story building with infinite extent of floors and ceilings (no walls). It was assumed that 20 percent of the energy input to the light was absorbed by the ceiling surface, 47.1 percent by the floor surface, and that 32.9 percent was transferred directly to the air of the enclosure. The floors were assumed to be of cinder concrete 10 in. thick with a thermal conductivity of 0.41 Btu per (hr) (sq ft) (F deg per ft) and a thermal diffusivity of 0.02 sq ft per hour.

In the solution of the partial differential equation of heat conduction in the unsteady state, the floor was divided into six slabs. A constant convection film coefficient of heat transfer of 0.25 Btu per (hr) (sq ft) (F deg) was assumed for the ceiling to air and 0.50 Btu per (hr) (sq ft) (F deg) for the floor to air. The rate of exchange of radiant energy between floor and ceiling was assumed to be 1 Btu per (hr) (sq ft) for each degree of temperature difference. The boundary equations for floor and ceiling surfaces were energy balances. Details of calculations are not included in this paper.

At the instant of first input of energy to the lights, the entire structure and the air were assumed to be at the same temperature, 75 F. It was assumed that conditioned air was supplied in such state and rate as to maintain a steady air temperature of 75 F.

Results of the calculation are shown in Fig. 1. One significant fact shown in the figure is that eight hours after the lights are turned on, the cooling load is only $\frac{6.9}{11.6}$ or 0.595 times the instantaneous rate of energy supply to the lights. It should be noted, however, that the operative temperature of the environment is increasing with time. Although the temperature of the air is constant, the mean radiant temperature of the enclosure increases as the temperatures of the floor and ceiling surfaces rise.

[1] Exponent numerals refer to References.

OCCUPANCY LOAD

Example 2: The cooling load resulting from human occupancy was studied in this specific case. The rate of sensible heat emission per occupant was assumed to remain steady at 200 Btu per hour, one-half of which was assumed to be transferred directly

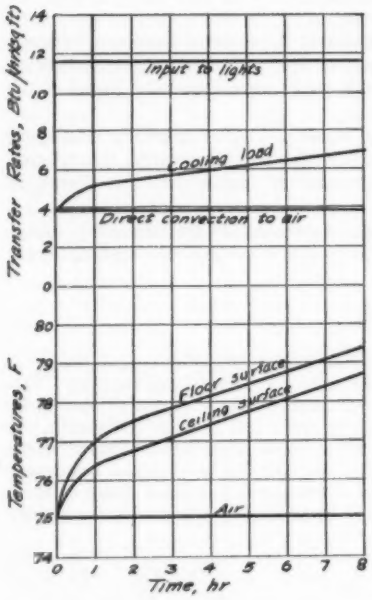

FIG. 1. TEMPERATURES AND HEAT TRANSFER RATES FOR *Example 1*

Multi-story building with 10-in. cinder concrete floors (no walls); Steady lighting load

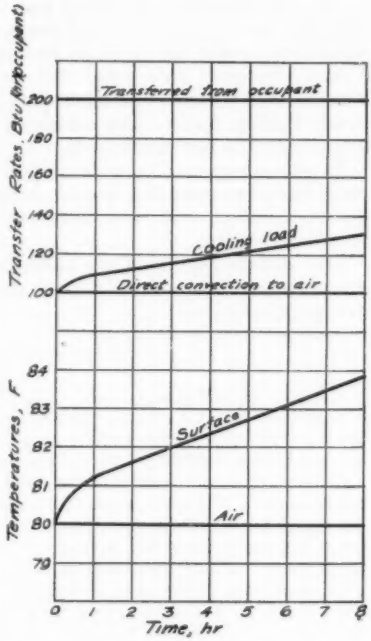

FIG. 2. TEMPERATURES AND HEAT TRANSFER RATES FOR *Example 2*

Steady load from human occupancy of enclosure similar to theater; Steady air temperature

to the air, while one-half was radiated to the surrounding surfaces of the enclosure. The temperature of the air was assumed to be held constant at 80 F by the supply of conditioned air and at the first instant of occupancy, the temperature of the entire structure was assumed to be 80 F. The area of the enclosing surface was assumed to be 20 sq ft per person, similar to conditions in a theater, and the enclosing material was assumed to have an average thickness of 6 in. with a volumetric specific heat of 30 Btu per (cu ft) (F deg). The temperature of the remote surface of the enclosure was assumed that of an adiabatic surface (no transfer of heat). The average value of the film coefficient of heat transfer by convection from enclosing surface to air was assumed to be 0.4 Btu per (hr) (sq ft) (F deg).

The results of the calculation are shown in Fig. 2. At the end of eight hours, the cooling load is only $\frac{130}{200}$ or 0.65 times the rate of emission of energy per occupant.

With the temperature of the enclosing surface rising as the temperature of the air remains constant, a steady condition of comfort would not be maintained; the operative temperature of the environment would rise.

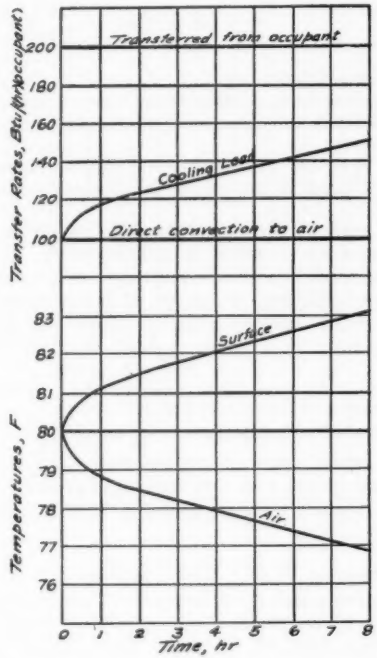

FIG. 3. TEMPERATURES AND HEAT TRANSFER RATES FOR *Example 3*

Steady load from human occupancy of enclosure similar to theater; Steady operative temperature

STEADY OPERATIVE TEMPERATURE

Example 3: This example is presented as a study of cooling load associated with the maintenance of a steady operative temperature rather than a steady air temperature. It was assumed that a steady operative temperature of the enclosure of *Example 2* might be maintained by keeping the sum of the air temperature and the temperature of the enclosing surface equal to a constant value. In other words, cooling was assumed to be provided in the amount necessary to lower the air temperature one

degree for each degree rise in temperature of the enclosing surface. The results of this calculation are shown in Fig. 3. At the end of eight hours, the cooling load in this case is $\frac{150}{200}$ or 0.75 times the rate of emission of energy per occupant.

From these examples and other considerations, the authors conclude that the true cooling load resulting from a steady source of energy either stored or transferred to the air may be represented by the following equation:

$$L = f_c E + R f_r E \quad \ldots\ldots\ldots\ldots\ldots (1)$$

where

L = cooling load, Btu per hour.
E = steady rate of energy emission from source of load, Btu per hour.
f_c = fraction of E that is transferred directly to air (principally by convection from source).
f_r = fraction of E that is radiated to and absorbed by enclosing surfaces.
R = retransfer factor, or the fraction of $f_r E$ which is transferred to the air.

To estimate true cooling load resulting from a steady source of energy requires a knowledge of f_c, f_r and R in addition to E. Current conventional methods of estimating cooling load are equivalent to assuming that $R = 1$. If at the instant of the beginning of energy emission from steady source the temperature of the entire storage material equals the air temperature, the true value of R depends (in a complex way) upon the following principal variables:

1. Time θ after beginning of emission of energy.
2. Average thickness x of enclosing and storing material.
3. Average thermal conductivity k of enclosing material.
4. Average volumetric specific heat C of enclosing material.
5. Average film coefficient h of heat transfer by convection from storage surface to air.

The relationship between R and these variables will never be represented by any equation sufficiently general to apply in all possible cases. In Appendix A limiting values of R are derived for a case of uniform storage. It is assumed that all energy emitted from the steady source is either stored or transferred to the air. In this case, the storing material is assumed to have an infinitely large thermal conductivity so that the temperatures at various points in the material are assumed to be the same at a particular time. As a result of this assumption the temperature of the enclosing surface at any time is lower than would actually exist and consequently yields a minimum value of R.

One limiting value of R is for the case where a steady operative temperature of the environment is maintained by reducing the air temperature as the surface temperature rises. Then, the minimum retransfer factor is

$$R_1 = 1 - \frac{1}{e^y} \quad \ldots\ldots\ldots\ldots\ldots (2)$$

where

e = Naperian base of logarithms

$y = \frac{2h\theta}{Cx}$

Another limiting value of R is for the case where a steady air temperature is maintained. Then, the minimum retransfer factor is

$$R_2 = 1 - \frac{1}{e^{y/2}} \quad \ldots \ldots \ldots \ldots \ldots \quad (3)$$

In the equation for y, the variables h, θ, C, and x are expressed in units that will make y dimensionless; commonly, h is in Btu per (hour) (square foot) (Fahrenheit degree); θ is in hours; C is in Btu per (cubic foot) (Fahrenheit degree); and x is in feet.

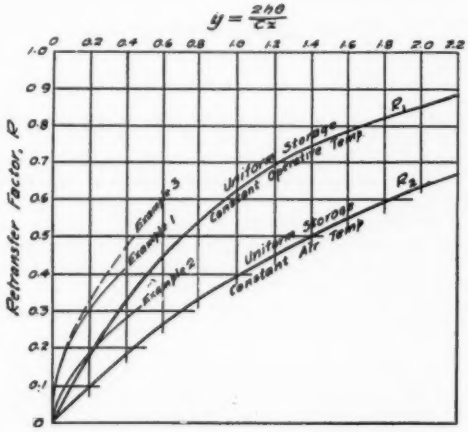
y = 2hθ / Cx
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2
1.0 0.9 0.8 0.7 0.6 0.5 0.4 0.3 0.2 0.1 0
Retransfer Factor, R
R1
R2
Uniform Storage Constant Operative Temp
Uniform Storage Constant Air Temp
Example 3
Example 1
Example 2

FIG. 4. CHART FOR ESTIMATING VALUE OF RETRANSFER FACTOR R

Values for the retransfer factors R_1 and R_2 are shown in Fig. 4. In this figure are also shown the actual retransfer factors found in *Examples 1, 2,* and *3*. A constant operative temperature was maintained in *Example 3* and a constant air temperature in *Examples 1* and *2*. The actual retransfer factors obtained in *Examples 1* and *2* lie above R_2 for any value of y, and the actual retransfer factors in *Example 3* lie above R_1. The reason for this is that uniform storage was not assumed in working the examples. Unsteady state conduction of heat through a material with finite conductivity and with a temperature gradient was going on. With these conditions, the surface temperature at a particular time reaches higher values than with uniform storage.

Fig. 4 is presented as a guide in estimating a reasonable value for the retransfer factor in any actual case. It is clear that this factor is significantly smaller than 1.0. The analysis does not show the effect of pre-cooling or of conduction of heat out of the enclosure. Obviously, smaller values of R may be used if the material of the structure is cooled before the beginning of energy emission, or if heat is conducted out of the enclosure.

Summarizing: In Equation 1 the value to be assumed for the retransfer factor R will be *low* for: pre-cooling; conduction of heat away from enclosure; high values of k, C, and x; and low values of h and θ.

Periodic Sources: Among the important periodic sources of cooling load are: (1) energy transferred through roofs and walls; (2) energy transferred through sunlit glass. Specific examples are presented to illustrate the difference between instantaneous rates of heat transfer from the room side of horizontal roofs and the true contributions to the cooling loads from such heat transfers.

In THE GUIDE, the instantaneous rate of heat transfer from horizontal roofs is estimated by means of the following equation:

$$\frac{q}{A} = U\,(t_m - t_i) + \lambda U\,(t_e^* - v_m) \quad \ldots\ldots\ldots\ldots \quad (4)$$

where

$\frac{q}{A}$ = instantaneous rate of heat transfer, Btu per (hour) (square foot).

t_m = average daily sol-air temperature for the locality, solar absorptivity of roof surface, and outdoor air film coefficient of heat transfer, Fahrenheit degrees.

t_i = constant indoor air temperature, Fahrenheit degrees.

t_e^* = sol-air temperature at a time earlier than the time for which the rate of heat transfer is being found by an amount equal to the time lag of the roof, Fahrenheit degrees.

λ = amplitude decrement factor with a value that depends upon the thickness and material of the roof.*

The values of λ are based on charts which use an outdoor air film coefficient of heat transfer of 4.0 Btu per (hr) (sq ft) (F deg) and an indoor surface coefficient of heat transfer by radiation and convection of 1.65 Btu per (hr) (sq ft) (F deg). If these same values are used in calculating the overall coefficient of heat transfer,

$$U = \frac{1}{0.856 + \sum \frac{L}{k}} \text{ Btu per (hr) (sq ft) (F deg)}$$

where

$\sum \frac{L}{k}$ = thermal resistance to conduction of heat of roof materials.

The corresponding value of the temperature of the room side of the roof is

$$t_c = t_i + 0.606\,\frac{q}{A}$$

When the temperature of the air in the enclosure is held constant, this calculated value of heat transfer may fail to represent the true cooling load from the roof at any particular time if any of the energy radiated from the room side of the roof is being stored in absorbing materials or conducted from the enclosure to any heat sink outside the building. Also subject to question is the

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* HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1949, CHAPTER 15, TABLE 12, p. 289.

value of 1.65 for the so-called "combined" surface coefficient of heat transfer. For the levels of temperature and the materials involved, the rate of energy transfer by radiation to the surfaces surrounding the ceiling is substantially $(t_c—t_s)$ Btu per hour per square foot per Fahrenheit degree, where t_s represents the average temperature of the absorbing surfaces. McAdams[3] recommends the following equation for surface coefficient of heat transfer by convection from warm ceiling to cooler air:

$$h = 0.2\ (t_c - t_i)^{0.25}\ \text{Btu per (hr) (sq ft) (F deg)}$$

There is no doubt that the convection transfer from the ceiling is a direct cooling load, but depending upon the value of the retransfer factor, only part of the energy radiated from ceiling to surroundings may be cooling load.

EFFECT OF HEAT STORAGE AND CONDUCTION

Example 4: This example shows how the cooling load may be far less than the instantaneous rate of heat transfer from the room side of the roof. This is caused by the combined effects of (1) storage of energy in absorbing materials, and (2) conduction of heat to a heat sink outside the enclosure.

In this example, the building consists of a single story with indefinite extent of roof and floor and no walls. All energy radiated from the ceiling is assumed to be absorbed by the floor. The assumed roof construction is 2 in. concrete and 1 in. rigid insulating board. The floor is assumed to be a 10-in. slab of cinder concrete ($k = 0.41$ Btu per (hr) (sq ft) (F deg per ft); thermal diffusivity = 0.02 sq ft per hour) directly on the ground; the earth side of this floor slab is assumed to remain at 55 F, and the temperature of the air is assumed to be constant at 80 F.

The temperature of the ceiling was calculated from sol-air temperature data in the conventional manner (solar absorptivity = 1.0; $\lambda = 0.85$; $U = 0.2$; time lag = 1.5 hours; locality, New York). The floor was divided into 5 slabs. The Schmidt method was used to determine the effects of heat conduction in the unsteady state. Results are shown in Fig. 5.

At the time of maximum true cooling load (2:30 p.m.), it is interesting to note the following:

$$1.65\ (t_c - 80) = 1.65\ (88.2 - 80)$$
$$= 13.5\ \text{Btu per (hour) (square foot)}$$

Conventional estimated load

$$= (88.2 - 80) + 0.2\ (88.2 - 80)^{1.25}$$
$$= 11\ \text{Btu per (hour) (square foot)}$$

True rate of transfer of heat from ceiling to air and floor

$$= (88.2 - 77.2) + 0.2\ (88.2 - 80)^{1.25}$$
$$= 13.8\ \text{Btu per (hour) (square foot)}$$

True cooling load, ceiling and floor convection

$$= 0.2\ (8.2)^{1.25} - 0.2\ (2.8)^{1.25}$$
$$= 2.0\ \text{Btu per (hour) (square foot)}$$

Rate of conduction of heat into ground

$$= 10.1\ \text{Btu per (hour) (square foot)}$$

Rate of storage of heat in floor slab

= 1.7 Btu per (hour) (square foot)

Example 5: For the purpose of presenting the effect of storage alone without any transfer of heat away from the enclosure, *Example 5* is chosen to represent another

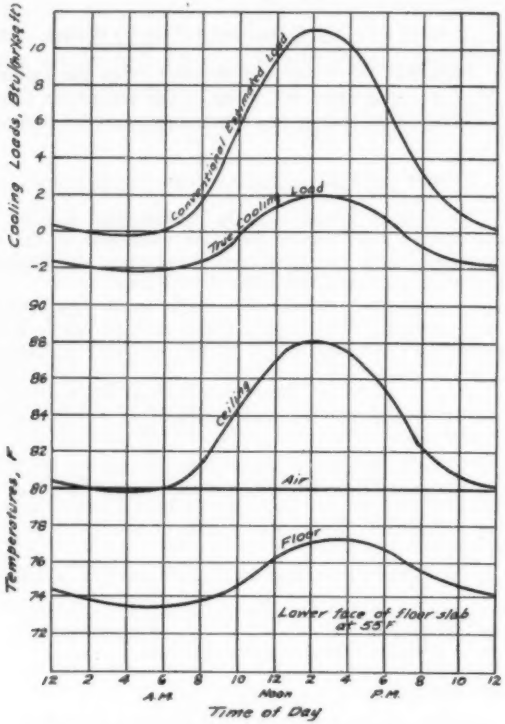

FIG. 5. TEMPERATURES AND COOLING LOAD FOR *Example 4*

Single-story building; 10-in. cinder concrete floor slab on ground

building, without walls, and with a roof slab of 4 in. cinder concrete and a floor slab of 6 in. cinder concrete. The temperature of the air is assumed constant at 80 F. The under side of the floor slab is assumed to be an adiabatic surface across which no heat is transferred. Sol-air temperatures and a solar absorptivity of 0.7 were taken for New York. In this example, the transfer of heat is studied entirely by means of the Schmidt method, and no previous calculations other than sol-air temperature are required. The results are shown in Fig. 6.

At the time of maximum cooling load (2:30 p.m.) the following instantaneous rates are of interest, such as the rate of:

Entry of heat into weather surface of roof = 23.6 Btu per (hour) (square foot).
Transfer of heat from ceiling to air = 12.8 Btu per (hour) (square foot).

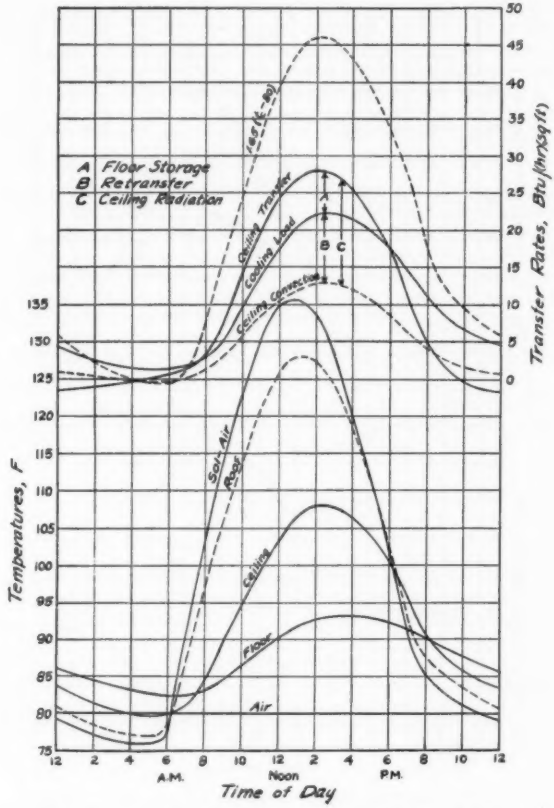

FIG. 6. TEMPERATURES AND TRANSFER RATES FOR *Example 5*

4-in. cinder concrete roof slab; 6 in. cinder concrete floor slab with adiabatic remote surface constant air temperature

Radiation of energy from ceiling to floor = 15.4 Btu per (hour) (square foot).
Storage of energy in roof material = —4.6 Btu per (hour) (square foot).
Retransfer of heat from floor to air = 9.3 Btu per (hour) (square foot).
True cooling load = 12.8 + 9.3 = 22.1 Btu per (hour) (square foot).
Rate of storage of energy in floor = 6.1 Btu per (hour) (square foot).

Ratio of retransfer from floor to radiation from ceiling $= \frac{9.3}{15.4} = 0.6$.

At the time of maximum cooling load, 60 percent of the radiation emitted by the ceiling is being retransferred to the air by the floor surface and 40 percent is

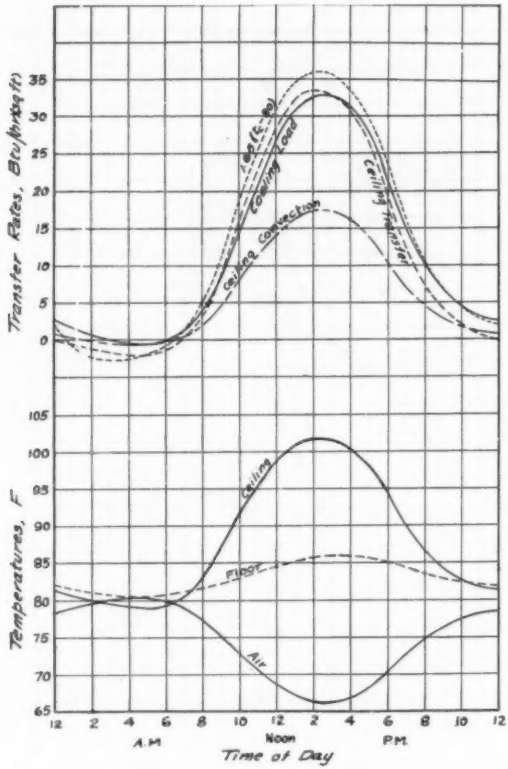

FIG. 7. TEMPERATURES AND HEAT TRANSFER RATES FOR *Example 6*

4-in. cinder concrete roof slab; 6-in. cinder concrete floor slab with adiabatic remote surface; constant operative temperature

going into storage in the floor. When the enclosure is in heat exchange relationship only with outdoor air and sun (as in this example), the heat transferred in 24 hours from the ceiling and floor to the air must be the same as the heat transferred from the ceiling to the air and floor in that time.

If, at some times, the cooling load is less than the instantaneous rate of transfer of heat from the ceiling, at other times this situation must be reversed. As Fig. 6 shows,

the retransfer factor is exactly 1.0 at two times of day, about 8 a.m. and 6 p.m. During the day, between 8 a.m. and 6 p.m., the retransfer factor is less than 1.0. During the evening, between 6 p.m. and 8 a.m., the retransfer factor is greater than 1.0.

Example 6: Although the temperature of the air is held constant at 80 F in *Example 5*, a steady operative temperature is not being maintained in the enclosure because of the varying temperatures of the ceiling and floor. *Example 6* is the same building as *Example 5* with a constant *operative temperature* of 80 F instead of a constant *air temperature* of 80 F. The operative temperature is assumed to be equal to the sum of one-fourth of the temperature of the ceiling, one-fourth of the temperature of the floor, and one-half of the temperature of the air. In other words, attention is given to the effect of mean radiant temperature upon comfort. The results are shown in Fig. 7. At the time of maximum cooling load (2:30 p.m.), the following instantaneous rates are of interest, such as rate of:

Entry of heat into weather surface of roof = 28.4 Btu per (hour) (square foot).
Transfer of heat from ceiling to air = 17.5 Btu per (hour) (square foot).
Radiation of energy from ceiling to floor = 16 Btu per (hour) (square foot).
Storage of energy in roof material = −5.1 Btu per (hour) (square foot).
Retransfer of heat from floor to air = 15.7 Btu per (hour) (square foot).
True cooling load = 17.5 + 15.7 = 33.2 Btu per (hour) (square foot).
Rate of storage of energy in floor = 0.3 Btu per (hour) (square foot).

$$\text{Ratio of retransfer from floor to radiation from ceiling} = \frac{15.7}{16} = 0.98.$$

In this example, the instantaneous cooling load is never significantly different from the instantaneous rate of heat transfer from ceiling to air and floor. In other words, the retransfer factor is never significantly different from 1.0.

CONCLUSIONS

Steady Sources of Energy Emission

1. The instantaneous contributions to the cooling load from steady sources of energy emission having appreciable components of radiation (such as lights, people, and appliances) may be significantly less than the instantaneous rates of energy emission.
2. Sufficient examples with simplifying assumptions have been presented to permit a reasonable estimate of the retransfer factor. It is suggested that Equation 1 and Fig. 4 be used to estimate cooling loads from steady sources.
3. Additional information that might be compiled to assist in the estimate of cooling loads from steady sources consists of the fraction of the energy emission in the form of convection and the fraction in the form of radiation.

Periodic Sources of Energy Emission

1. The instantaneous contributions to the cooling load from sources of periodic energy emission like roofs, walls, and sunlit glass may be less than the instantaneous rates of energy emission at the time of maximum cooling load.
2. The instantaneous cooling loads may be appreciably less than the instantaneous rates of energy emission under the following conditions: (*a*) when pre-cooling or after-cooling is used, (*b*) when a constant air temperature is maintained, and (*c*) when any appreciable portion of the structure is in heat exchange relationship with a relatively cool heat sink, like the ground, in addition to being in heat exchange relationship with the sun and outdoor air.
3. If a steady operative temperature is maintained by lowering the temperature of the air as the mean radiant temperature of the enclosure rises, the instantaneous

maximum cooling load may not be significantly less than the instantaneous rate of energy emission at the time of maximum load.

4. More study should be given the problem of estimating cooling loads from sources of periodic energy emission.

APPENDIX A

The following symbols are used:

E = steady rate of energy emission, Btu per (hour) (square foot of floor space).
L = actual cooling load at any time, Btu per (hour) (square foot of floor space).
A = area of storage surface per square foot of floor, square feet.
f_c = fraction of E that is transferred directly to air.
f_r = fraction of E that is radiated to and absorbed by enclosing surface.
t = temperature of storing surface at any time, Fahrenheit degrees.
t_a = temperature of air at any time, Fahrenheit degrees.
x = average thickness of storage material, feet.
Ax = volume of storage material per square foot of floor, cubic feet.
C = volumetric specific heat of storage material, Btu per (cubic foot) (Fahrenheit degree).
h = average film coefficient of heat transfer by convection from storage surface to air, Btu per (hour) (square foot) (Fahrenheit degree).
θ = time after beginning of steady energy emission, hours.

Special assumptions: all energy radiated from steady source is either stored or convected to room air; no temperature gradient in materials storing energy. The latter assumption, the equivalent of assuming the storage material to have an extremely large thermal conductivity, may be regarded as a limiting case only.

Energy balance

$$f_r E = hA\,(t - t_a) + C\,Ax\,\frac{dt}{d\theta} \quad \ldots\ldots\ldots\ (A1)$$

$$\int_0^{\theta} d\theta = CAx \int_{t_o}^{t} \frac{dt}{f_r E - hA\,(t - t_a)} \quad \ldots\ldots\ldots\ (A2)$$

Special cases of integration: Case 1. Assume that comfort is maintained by having the temperature of the air decrease one degree for each degree of increase in the temperature of the surface of the enclosure.

Comfort equation

$$t + t_a = K \quad \ldots\ldots\ldots\ (A3)$$

where K is a constant.

Integration of Equation $A2$ under the conditions imposed by Equation $A3$ gives

$$\int_0^{\theta} d\theta = -\,CAx \int_{t_{ao}}^{t_a} \frac{dt_a}{f_r E - hA\,(K - 2\,t_a)}$$

and

$$t_a = \frac{2hA\,t_{ao} - f_r E\,(e^y - 1) + hAK\,(e^y - 1)}{2\,hA\,e^y} \quad \ldots\ldots\ (A4)$$

where

$$y = \frac{2h\theta}{C\,x} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (A5)$$

The contribution to the cooling load from the energy radiated from the steady source is

$$L_r = hA\,(t - t_a) = \frac{hA\,(K - 2t_{ao}) + f_r E\,(e^y - 1)}{e^y} \quad \ldots\ldots \quad (A6)$$

If it be assumed that, at $\theta = 0$, the uniform temperature of the storage material is the same as that of the air at that time (or $t_o = t_{ao}$), and if the comfort equation is satisfied at that time,

$$L_r = \frac{f_r E\,(e^y - 1)}{e^y} \quad \ldots\ldots\ldots\ldots \quad (A7)$$

The ratio of this instantaneous contribution to the cooling load to the instantaneous rate of energy emission by *radiation* from the steady source is

$$R_1 = \frac{L_r}{f_r E} = 1 - \frac{1}{e^y} \quad \ldots\ldots\ldots\ldots \quad (A8)$$

Case 2. Assume that the temperature of the air in the enclosure is held constant. Integration of Equation *A2* under the new conditions gives

$$t = \frac{hAt_o + f_r E\,(e^{y/2} - 1) + hA\,t_a\,(e^{y/2} - 1)}{hA\,e^{y/2}} \quad \ldots\ldots \quad (A9)$$

The contribution to the cooling load from the energy radiated from the steady source is

$$L_r = hA\,(t - t_a) = \frac{hA\,(t_o - t_a) + f_r E\,(e^{y/2} - 1)}{e^{y/2}} \quad \ldots\ldots \quad (A10)$$

If it be assumed that, at $\theta = 0$, the temperature of the storage material is the same as that of the air at that time (or $t_o = t_a$),

$$L_r = \frac{f_r E\,(e^{y/2} - 1)}{e^{y/2}} \quad \ldots\ldots\ldots\ldots \quad (A11)$$

The ratio of this instantaneous contribution to the cooling load to the instantaneous rate of energy emission by *radiation* from the steady source is

$$R_2 = \frac{L_r}{f_r E} = 1 - \frac{1}{e^{y/2}} \quad \ldots\ldots\ldots\ldots \quad (A12)$$

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DISCUSSION

LESTER T. AVERY, Cleveland, Ohio (WRITTEN): Mr. Herb Heisterkamp poses the question as to the ability of walls, particularly insulated walls, to hold the water vapor accumulated during the cool night season which acts as an evaporative cooler during the warm day. He suggests that this storage ability for water and its re-evaporation has a lot to do with the very low cooling loads required for residential work.

The points raised by the authors on heat gains figured for cooling loads are very much in order. We have suspected that in figuring the full load as to lights, people and other fluctuating loads, particularly sun load, that we did not have all the maximum loads occurring at one time. The compensating error in figuring loads is, of course, the general one that we talk about 80 F and 50 percent relative humidity for inside design when it is 95 F, or 74 ET outside. We actually carry much lower temperatures, between 69 ET and 71 ET and below 50 percent relative humidity practically all the time so that we are always carrying systems considerably below the design guaranteed point.

On large installations where there is a substantial building construction load compared with the other loads, for example in a home or in an office where there is not a great internal load, we have observed that equipment seems to be considerably over-sized when figured according to the regular procedure. This paper gives one reason for it. There is another reason advanced by Mr. Heisterkamp that is worth bringing out here.

Mr. Heisterkamp contends that the building materials, particularly any insulating material such as sugar cane fiber or rock wool absorbs water vapor during the night cooling period which acts as a reservoir of water vapor from which we gain *evaporative cooling* effect when the heat comes on during the day. This is particularly true in homes exposed to sunlight where the insulating materials used in the east, south and west walls and the roof, all can absorb water vapor during the humid evening and night hours and later give up its stored cooling effect by evaporation when the sun strikes it during the day. This accounts for the fact that the walls in these residences remain cool for a long time, much longer than would be the estimated time lag. The quantity of water vapor so absorbed is unknown.

Mr. Heisterkamp suggests that this should be investigated in the Society's Laboratory. The cooling load required for residential conditioning can be done with very modest size equipment if it is planned to take advantage of this evaporative cooling effect from absorbed moisture in the walls. If along with this we consider the storage capacity of the materials for absorbing of radiant heat from lights and the human body, we will find it possible to do much more economical cooling and provide satisfactory results.

We are indebted to the authors for this fine paper which brings out some much appreciated points in the study of air conditioning loads.

R. S. DILL, Washington, D.C. (WRITTEN): It is very fortunate that steady-state heat flow problems are by nature simple and capable of representation with simple formulae since this makes it possible, as a common practice, to estimate required capacities for heating and air conditioning equipment with some degree of precision. To improve that precision it is desirable to take heat capacities of materials as well as thermal conductivities into account, but the complexity of this problem makes it doubtful that it will ever be done with the facility and certainty with which steady-state problems are handled in everyday practice. The present paper is regarded as a worthwhile effort to simplify the subject. I agree that more study should be given the problem of estimating cooling loads from sources of periodic energy emission and perhaps further effort will yield practical computation methods of approximations

which will facilitate design and result in considerable savings by permitting choice of smaller or more reasonably sized air conditioning equipment.

If we could live in dwellings or occupy buildings with walls about 10 ft thick, our air conditioning loads might diminish almost to the vanishing point. This being impossible on account of the support problem, it appears reasonable to take advantage, when possible, of the heat capacity of the structures we do utilize which, in heat capacity, are somewhere between the building with 10-ft walls and a tent.

It is interesting to speculate on whether or not it is feasible and desirable to put the masonry and structural masses of our buildings on the inside rather than on the outside of the insulating layer and thus increase the effective heat capacity of the structure to take advantage of the phenomenon which is the subject of this paper.

JOHN EVERETTS, JR., Philadelphia, Pa. (WRITTEN): I wish to take this opportunity to compliment the authors for their fortitude in attacking a problem which is undoubtedly the most controversial subject of air conditioning.

There are three points of discussion which I would like to bring out, the last of which will undoubtedly require additional research and analysis.

The first point is in reference to *Example 3,* which is a study of steady load from human occupancy of enclosure similar to a theater, with steady operative temperatures. The authors have assumed a radiation release of 100 Btu per hour per person. Fig. 3 shows that at the end of an 8-hr period, the air temperature is maintained at 77 F and the surface temperature of the enclosure is 83 F. According to THE GUIDE, the heat release through radiation and convection at 77 F dry bulb, is 240 Btu per hour and the surface temperature of a clothed person is 86 F.

With 86 F surface temperature of people and an 83 F surface temperature of an enclosure, the calculated radiant heat emission is only 40 Btu per hour instead of the assumed 100, leaving 200 Btu per hour per person as heat release to the air at 77 F dry bulb. This, of course, theoretically increases the cooling load beyond that indicated in the example. If the radiant heat release from people under this condition is lower than that used in the example, the surface temperature of the enclosure will not be increased as much and the air temperature will not necessarily be decreased to 77 F to maintain a steady operative temperature. Therefore, the surface temperature of the enclosure will not be quite as high as shown in Fig. 3 and the air temperature will not be quite as low. Obviously, a condition of equilibrium will be reached following a cooling load somewhat higher than that indicated in Fig. 3 and an air temperature somewhere between 77 and 80 F.

The second point of discussion is that where there is a high concentration of people, such as a theater or restaurant, the radiation from the people will be partly to each other and partly to the chairs or seats, so that all the radiation cannot be assumed to be picked up by the surfaces of the enclosure. It is obvious that under such conditions a greater proportion of this radiant heat will be converted into convection, thus increasing the cooling load.

The third point of discussion, which will need additional study, is the question of humidity. J. C. Albright, in his study of long-term storage of ordnance material, brings out the interesting fact that the hygroscopic materials which are contained within a reasonably tight enclosure will produce what he terms a lag loop in the atmospheric conditions surrounding these materials. If the temperature in the enclosure is increased, water vapor is released from the materials which increased the dewpoint temperature and moisture content of the surrounding air. If the temperature again decreases, the moisture is absorbed back into the materials which results in a hysteresis effect which is a function of temperature variation and time. This effect can be appreciable, particularly in a department store containing many tons of hygroscopic materials and which may be shut down over the weekend. Preliminary calculations of one medium sized department store indicates that a 10 deg rise in temperature over the weekend will release approximately 1,500,000 Btu

of latent heat which must be removed from the space on a Monday morning pull down along with latent infiltration. This increased pull down load will result in a higher ambient dry bulb in the space which will reduce the sensible cooling load and increase the heat storage in the surfaces of the enclosure. This will require a greater period of time to reach an equilibrium condition, so that the calculations for one 8-hr day do not show the true picture of actual cooling load involved.

I readily admit that the introduction of latent load makes this problem much more complex. However, as the condition exists, there is nothing else to do but recognize it.

As the authors of this paper have made such an excellent start in this study, I sincerely hope that they will continue with this work so that we may be able to more accurately determine the cooling load in air conditioning problems.

J. DONALD KROEKER, Portland, Ore. (WRITTEN): As a presentation of *methods* of analyzing the magnitudes of known discrepancies between sources of instantaneous heat gains and cooling loads, the paper is an extremely valuable contribution to air conditioning engineering. Lucid and accurate, it makes generous use of charts for typical conditions on clearly defined assumptions. These charts will be valuable as guides in analyzing specific analogous situations.

It will be recognized that the paper is limited to discussion of methods, that it is not intended for application of values directly, and that it discusses only treatment of sensible heat gains. It furnishes the engineer with tools to assist in estimating cooling loads and will narrow the scope of research the Society must undertake to evaluate certain factors needed by the engineer to preclude over-design of cooling systems, which results from computation of cooling loads without proper evaluation of the extent to which heat gains are contributory to loads.

C. S. LEOPOLD, Philadelphia, Pa. (WRITTEN): Up to a few years ago, it was tacitly assumed that to utilize thermal storage one could either allow room temperature to rise or precool the structure. We now know that in air cooling systems, with thermal loads consisting in part of radiation, storage is almost inevitable and plays a large part in the operation of most air conditioning installations.

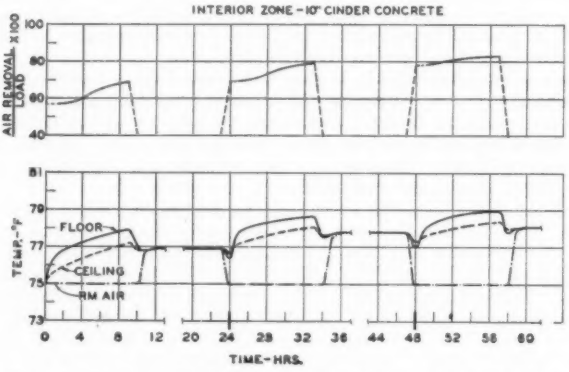

FIG. A. ANALOGUE SOLUTION OF PROBLEM FOR 1 HR PRECOOLING AND 1 HR AFTERCOOLING

Thermostat Control: 2.7 watts per square foot; 6.7 percent Radiant; 10 percent IRT 1 Hr precool 1 Hr aftercool.

Both in this paper and in my paper* on the hydraulic analogue the analysis has been confined to one day in which the air temperature and the structure temperature have been considered uniform and equal at zero time. When a real system is operated to maintain a constant air temperature for a day's operation, it is apparent that the structure will be at a temperature above the design room air temperature on the succeeding day. Fig. A shows the results of an analogue solution of a problem, in which there was included one hour of precooling and one hour of after-cooling with the room air maintained constant at the design temperature. For each succeeding day the structure is at a higher temperature and for each day there is an increasing requirement for cooling approaching equilibrium.

Precooling and after-cooling depend primarily on convection. The effect can be accelerated by allowing the room air temperature to fall. Even under these conditions, the cooling is relatively slow and long periods are required to produce an

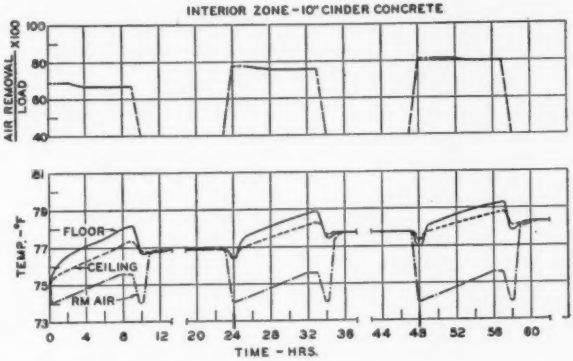

FIG. B. EFFECT OF AUTOMATIC TEMPERATURE CONTROL ON COOLING REQUIREMENTS

Thermostat Control: 1.6 deg variation; 3.7 watts per square foot; 50 percent Radiant; 10 percent IRT 1 Hr precool 1 Hr aftercool.

important effect. Precooling normally does not mean that the average temperature of the structure need be reduced below room air temperature but, frequently, means that its excess above room temperature is reduced. This is of practical significance since the building can be satisfactorily precooled by lowering the room air temperature and restoring thermostatic control shortly before its contemplated use without having the deleterious effect of starting with a cold room. In one field application, a considerably undersized store installation gave satisfactory results with approximately eight hours of after-cooling. The midpoint internal temperature of a thick ceiling construction was only reduced from 80 F to 79 F by the after-cooling and the midpoint, in general, tended to be approximately 3 deg higher than the mean air temperature.

With some types of automatic temperature control which act in response to room air temperature, an instrument drift to maintain temperature is to be anticipated. The effect of a total variation of ± 0.8 deg is shown in Fig. B, which indicates

* Hydraulic Analogue for the Solution of Problems of Thermal Storage, Radiation, Convection and Conduction, by Charles S. Leopold (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 389).

that the drift of this amount results in practically constant cooling requirements for any one day and a rise in cooling requirements from day to day. In a further series of experiments on the analogue, it was shown that if a ± 1.5 deg variation was permitted, the afternoon load was considerably less than the morning load, which may in part indicate how under-capacity systems sometimes meet the afternoon wet bulb peak.

This paper compares the results of maintaining constant operative temperatures with constant room air temperature. In analyzing this problem, there are several factors to be considered; the first is that thus far most of the discussion has centered on empty segments of an interior zone, a case in which there is a total of 2 sq ft of construction surface for 1 sq ft of floor surface. In the case of a cube, we would have 6 sq ft of structure for 1 sq ft of floor space. Surface temperatures would be expected to rise approximately one-third of the rise predicted for the infinite zone although the percent of storage might be approximately the same. This limited rise of surface temperature would result in a smaller correction for mean radiant temperature. To a degree, even in the interior zone, beams, columns, desks, filing cabinets, and other paraphernalia would tend to reduce the change in mean radiant temperature without proportionately reducing the storage effect. The second point to be considered is the varying comfort effect of mean radiant temperature, depending on whether or not the system is operated close to the optimum. The percentage effect on the number of individuals experiencing discomfort on an effective temperature change of 1 deg, from 70.5 to 71.5 F, is approximately 1 percent; from 71 to 72 F, 3 percent; whereas the effect from 73 to 74 F is 15 percent, a quantity which must be given serious consideration. In our American practice, we now know that very little departure from the optimum is permissible which, in a way, is fortunate since we can afford a moderate departure in mean radiant temperature without appreciable comfort effect.

I question the assumptions leading to the analysis of the storage effect of the radiation from a person because clothed body temperature is in the range where the total radiation is markedly affected by the change in wall temperature and, in the theater, I question the figures on radiation because people are seated and, also, to a large extent because their radiation in a horizontal direction is blocked. The analysis of Fig. 4 could be made to more nearly approach practice by assuming a higher ground temperature or by including a deep section of earth below which the temperature could be assumed constant at 55 F.

The authors have pointed out the limitations of their mathematical analysis. This analysis is valuable as an approximate mathematical approach but should be used with considerable caution as the assumption of infinite conductivity of the structural materials tends toward over-optimistic conclusions. The use of the concept of infinite conductivity also masks the effect of independent radiant transfer (IRT) and can add a sizable error in the retransfer factor as convection upward would be approximately twice the value for convection downward.

Although the examples in this paper contain many approximations, the overall result is of importance as a contribution to our knowledge of the general nature of the problem of heat storage, and particularly of the storage limits.

J. N. LIVERMORE, Detroit, Mich. (WRITTEN): This paper points again to a most serious blind spot in one of our profession's most important fields of working information, namely, the method of calculating a cooling load. Strangely enough, we can find quite precisely the rate at which heat enters an air conditioned enclosure, but we can make only the poorest sort of guess as to how rapidly that same heat reaches the cooling apparatus.

The authors are to be complimented on a very clear statement of a very complex problem. Their estimates of ranges of magnitude of error possible in estimating apparatus load from a radiant heat source, should attract wide attention in the

application engineering field. Their work should also prove valuable to others who will undertake research on this subject.

The authors refer to and use the term *constant operative temperature,* a condition which is not approached in actual practice. While I grant that all the examples shown are hypothetical, I think it should be emphasized that *constant operative temperature,* that is, a building in which the room dry bulb temperature is adjusted for changes in mean radiant temperature, is a rarity, indeed. Most systems operate either automatically or manually to maintain a constant air temperature.

J. P. STEWART, Syracuse, N.Y. (WRITTEN): The authors have shown that under favorable conditions a very large quantity of heat gain may be stored in an air conditioned space. The percentage stored seems to be even more than previously thought possible. I would very much like to see this excellent theoretical analysis followed soon by carefully controlled experimental work in the laboratory. This is a real awakening to the possibility of using smaller equipment and still doing a good job of cooling.

TABLE A. NEW CALCULATION RESULTS *(Example 2A)*

TIME, HR	TEMPERATURE OF ENCLOSING SURFACE, F	COOLING LOAD, BTU PER HR
0	80.00	100
1	80.06	109
2	80.12	113
3	80.14	116
4	80.16	119
5	80.18	122
6	80.21	125
7	80.23	128
8	80.25	130

W. E. ZIEBER, York, Pa. (WRITTEN): The TAC on Cooling Load, following discussion of the factors affecting cooling load, invited Professor Mackey to present a discussion on the general subject with the thought that such a discussion would be of great value to the Society. I believe the thinking of the Committee regarding the various phases involved has been covered very well by Professor Mackey. I know the TAC would be pleased if this work would incite discussion for the purpose of broadening the Committee's thinking and thereby allow its members to arrive at a successful conclusion, whether through experimental work or otherwise. The purpose is to establish a method of determining satisfactorily the cooling loads arising from various types of radiant heat sources and thereby help the engineers who calculate loads and design air conditioning equipment.

AUTHORS' CLOSURE: The authors are grateful to Mr. Leopold for his discussion and particularly for the data obtained from the hydraulic analogue showing the effects of operation on successive days. Mr. Leopold properly questions the assumption of *Examble 2,* where the rates of heat transfer from each occupant by radiation and convection were assumed to remain constant with time even though the temperature of the enclosing surface increased about 3.9 deg in 8 hr. To answer this criticism, *Example 2* has been recalculated with an assumed area of enclosing surface of 300 sq ft per occupant in place of 20 sq ft; other conditions were assumed to remain the same. The results of this new calculation, *Example 2A,* are given in Table A.

The temperature of the surface rises but little in 8 hr, and the assumption of no change in the rates of heat transfer by radiation and convection may now be justified. The importance of the new example is that the cooling loads and retransfer factors

TABLE B. COMPARISON OF RESULTS OBTAINED IN *Examples 4 and 4A* (AT 2:30 P.M.)

	EXAMPLE 4	EXAMPLE 4A
Temperature of ceiling, F	88.2	88.2
Temperature of room surface of floor slab, F	77.2	81.2
Conventional estimated load, Btu per (hr) (sq ft)	11.0	11.0
True rate of heat transfer from ceiling to air and floor, Btu per (hr) (sq ft)	13.8	9.8
True cooling load (ceiling and floor convection), Btu per (hr) (sq ft)	2.06	3.26
Rate of conduction of heat into ground, Btu per (hr) (sq ft)	10.10	4.50
Rate of storage of heat in floor slab, Btu per (hr) (sq ft)	1.64	2.04
Retransfer factor[a]	−0.065	+0.069

[a] The retransfer factor is increased but not by a large amount.

at a particular time are not changed from the results obtained in the original case.

As Mr. Leopold points out, the temperature 55 F assumed for the isothermal surface of the floor slab in contact with the ground in *Example 4* is too low to be of great practical significance. To answer this criticism, the authors have recalculated *Example 4* for a temperature of this surface of 70 F *(Example 4A)*. A comparison of the results obtained in *Examples 4* and *4A* at 2:30 p.m. are given in Table B.

Mr. Livermore points out that the most common method of control of air conditioning installations is to maintain a constant air temperature, not a constant operative temperature. The purpose of the authors in introducing operative temperature in some of the examples is to show that cooling loads and retransfer factors are higher if any attempt is made to reduce air temperature in order to compensate for rise in mean radiant temperature. In his discussion, Mr. Leopold has indicated the comfort tolerances that are involved.

The complexity of problems in unsteady state heat transfer is accented in Mr. Dill's discussion. From considerations of heat transfer and storage effects, alone, cooling loads may be reduced by the use of low-density materials near the weather side of a structure and high density materials near the room side.

The authors agree with Mr. Kroeker and Mr. Stewart that more data are necessary on the relation between cooling load and instantaneous rates of heat transfer, particularly for unsteady sources such as glass. Both additional analysis and experimental work would be helpful.

Mr. Avery and Mr. Everetts mention the behavior of hygroscopic materials and the cyclic nature of the effect of changes in moisture content of such materials of construction. No analysis of latent loads is contained in this paper, but the effects of storage on such loads may be significant.

Mr. Everetts questions the assumptions made in *Example 3*. This comment is similar to that of Mr. Leopold's criticism of *Example 2*. With an increase in the area of enclosing surface per occupant, the rise in temperature of the enclosing surface would be reduced and the assumption of steady rates of heat transfer by radiation and convection from the occupant could more readily be justified. It is doubtful, however, that the value of the retransfer factor would be changed significantly, and this is the important result. Any increase in convection from a source, and a corresponding decrease in radiation, tends to make storage effects less important and causes the cooling load to be a greater fraction of the rate of energy emission from source even for the same value of the retransfer factor.

The authors appreciate the considerable additional information which all of those who discussed this paper have contributed to the analysis of a complex problem.

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No. 1374

SOLAR ENERGY TRANSMITTANCE OF EIGHT-INCH HOLLOW GLASS BLOCK

By GEORGE V. PARMELEE* AND WARREN W. AUBELE**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

A CONTINUING program of research to develop design heat flow data for glass in its various architectural forms has been carried on since 1944 at the A.S.H.V.E. Research Laboratory under the direction of the Committee on Research and the TAC on Heat Flow Through Glass.† Previous papers[1,2] have given solar energy transmittance data for single sheets and various combinations of two sheets of flat glass. This paper presents similar data on five common patterns of eight inch glass block.

The problem of heat flow through a glass block panel to or from an enclosure is one of great complexity with respect to both heat conduction through the panel and the transmittance of short wave length radiant energy. Heat conduction through the panel is affected by outdoor and indoor boundary conditions, of which solar energy is one. The solar energy transmitting characteristics are affected by the size of the individual block, the face patterns, the absorption characteristics of the glass, the shading effect of the mortar joints, and the energy distribution of the incident solar radiation.

The principal purpose of this research report is to present solar energy transmitting characteristics. Data on heat conduction through the panel under steady state natural weather conditions are given. The problem of heat flow from the inside surface of the panel to the indoor surroundings as it is related to solar irradiation is considered only to the extent of giving approximate values of

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† Technical Advisory Committee on Heat Flow Through Glass: R. A. Miller, *Chairman;* A. B. Algren, A. H. Baker, F. L. Bishop, Jr., J. E. Frazier, J. S. Herbert, E. H. Hobbie, L. K. Jones, C. O. Mackey, F. W. Preston, W. C. Randall, C. A. Richardson, Vic Sanders, H. B. Vincent, G. B. Watkins, F. C. Wood.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Minneapolis, Minn., June 1949.

the magnitude and time of occurrence of peak heat gain. Further experimental work and analysis are required before a complete solution of this problem can be presented, especially with regard to relating instantaneous heat gains to system load.

EXPERIMENTAL PROCEDURE

The Calorimeter

Except for some minor changes, the calorimeter used in these studies and the instrumentation were essentially as described in some detail in previous

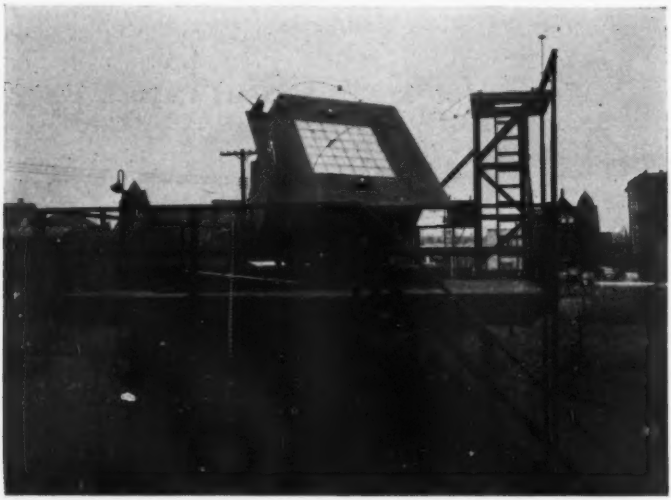


FIG. 1. CALORIMETER WITH GLASS BLOCK PANEL IN PLACE

reports.[1,2,3] Fig. 1 shows a glass block panel in place for test. In February, 1949, a convection-compensated radiometer was installed to measure the long wave length radiation from the ground surroundings and the sky that is incident on the panel area.

Test Panels and Temperature Measurements

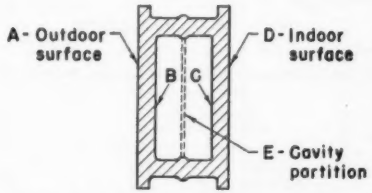
The blocks were set in wood frames using a sand-cement mortar to which a small amount of lime was added to improve consistency. Standard anchors and wall ties were set in the joints at intervals designated by the manufacturers. The description of the panels appears in Table 1.

Measurement of the panel surface temperature was an essential part of the experimental technique. The panel area was divided into four squares of equal

[1] Exponent numerals refer to References.

area. A single centrally located block in each of two diagonally located areas was selected for thermocouple installation. On each of these blocks, five thermocouples were cemented into shallow grooves cut into the block. Two other couples were located in the mortar joint adjacent to the block. The paralleled reading of 10 couples in the block and a similar observation of four couples in the mortar joint, weighted for relative area of glass and exposed mortar joint, gave the average panel surface temperature.

TABLE 1. DESCRIPTION OF GLASS BLOCK PATTERNS



Elevation Section of Hollow Glass Block to Indicate Location of Surface Patterns

Type I —Smooth Face
- A, D: Smooth
- B: Wide vertical ribs or flutes
- C: Wide horizontal ribs or flutes
- E: None

Type II —Semi-Light Diffusing
- A, D: Narrow vertical ribs or flutes
- B, C: Etched or stippled
- E: None

Type III —Light Diffusing
- A, D: Narrow vertical ribs or flutes
- B, C: Etched or stippled
- E: Glass fiber screen

Type IV —Light Diffusing
- A, D: Close pitch deep horizontal corrugations
- B, C: Vertical light diffusing prisms
- E: None

Type V —Light Directing
- A, D: Close pitch deep vertical corrugations
- B, C: Horizontal Light directing prisms
- E: None

RESULTS OBTAINED

Spectral Transmittance Characteristics

Flat polished samples of glass made from the same melts that were used to manufacture the glass block on which the heat flow studies were made were subjected to spectrophotometric tests. With these data transmittances for normally incident solar radiation were computed for two typical samples. The results are given in Table 2, part A. The average value of KL (the product of absorption coefficient K, per inch, and thickness, L, inches) for each sample was determined and used to compute the KL and the transmittances for a ¼ in. thickness. The transmittance, τ, is related to KL, as follows:

$$\tau = e^{-KL}/1.094$$

where

e = the Naperian base of natural logarithms

Part B of the same table gives the results of pyrheliometer tests made on samples taken from the blocks themselves.

Transmittance of Diffuse Solar Radiation

Table 3 lists the diffuse radiation transmittance of the five glass block panels tested. These were obtained by means of the calorimeter apparatus, and are

TABLE 2. TRANSMITTANCES OF GLASS BLOCK SAMPLES FOR NORMALLY INCIDENT SOLAR RADIATION

A. Computed from Spectro-photometric Data for Spectral Range 0.30 — 2.10 Microns[a]

SAMPLE	THICKNESS IN.	VALUES FOR SAMPLE TESTED		VALUES ADJUSTED TO ¼-IN. THICKNESS	
		Transmittance	Avg. KL	Avg. KL	Transmittance
A	0.496	0.870	0.0491	0.0247	0.892
B	0.494	0.816	0.1232	0.0622	0.859

B. Measured in Sunlight with Smithsonian Silver Disk Pyrheliometer

SAMPLE	DATE OF TEST	THICKNESS IN.	VALUES FOR SAMPLE TESTED		VALUES ADJUSTED TO ¼-IN. THICKNESS	
			Transmittance	Avg. KL	Avg. KL	Transmittance
A-1	7-15-48	0.166	0.89	0.022	0.033	0.89
A-1	8- 9-48	0.166	0.88	0.037	0.057	0.86
A-2	8- 9-48	0.215	0.88	0.036	0.039	0.88
B-1	7-15-48	0.292	0.87	0.053	0.046	0.87

[a] Based upon Moon's[4] energy distribution for air mass = 2; ordinates at 0.05 micron intervals.

averages of no less than six tests conducted at temperatures such that heat leakage corrections were at a minimum.

Transmittance of Direct Solar Radiation

To determine these values fixed angle tests were run in such a way as to determine effects of the mortar joint. One set of tests was run with the panel in such position that, at all times, the sun's rays lay in a plane which was perpendicular to the panel and parallel to the vertical mortar joints. This is referred to as Case I. In these tests, the shading effect was due only to the horizontal mortar joints. In a second set of tests, referred to as Case II, the sun's rays lay in a plane which was perpendicular to the panel, but parallel to the diagonal line across the panel. Here, both vertical and horizontal mortar joints shaded the glass opening. The results of these tests are shown in Fig. 2.

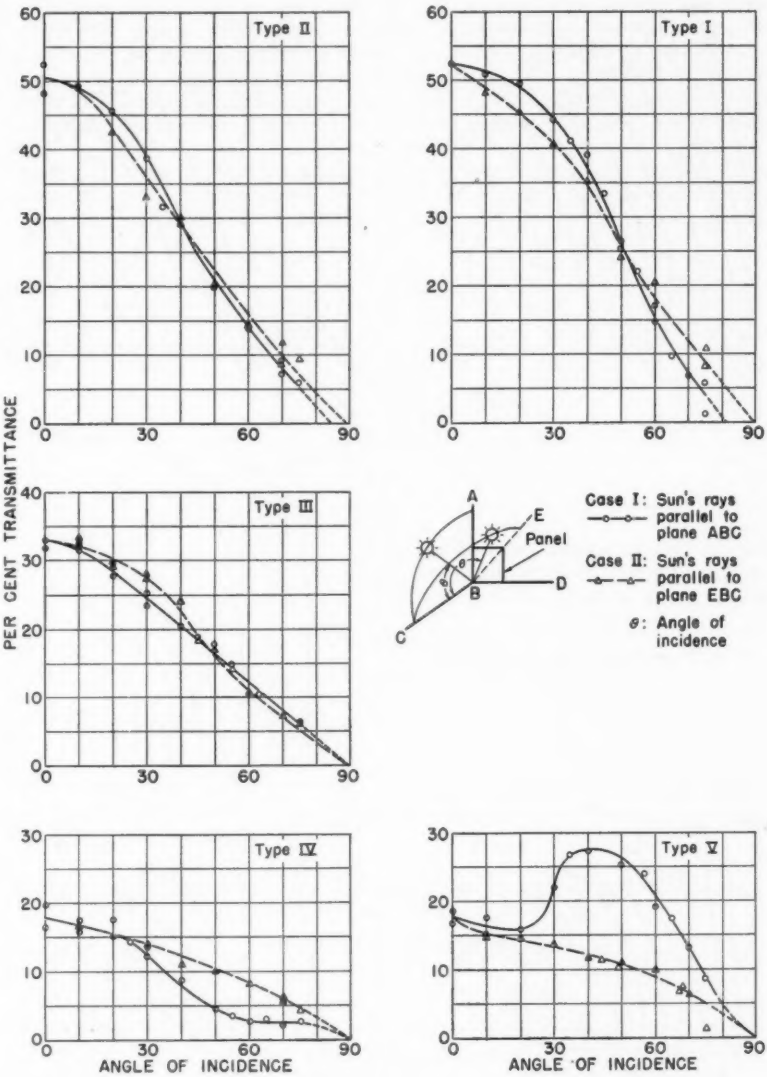


FIG. 2. TRANSMITTANCE OF DIRECT SOLAR RADIATION BY 8-IN. GLASS BLOCK PANELS IN VERTICAL WALLS

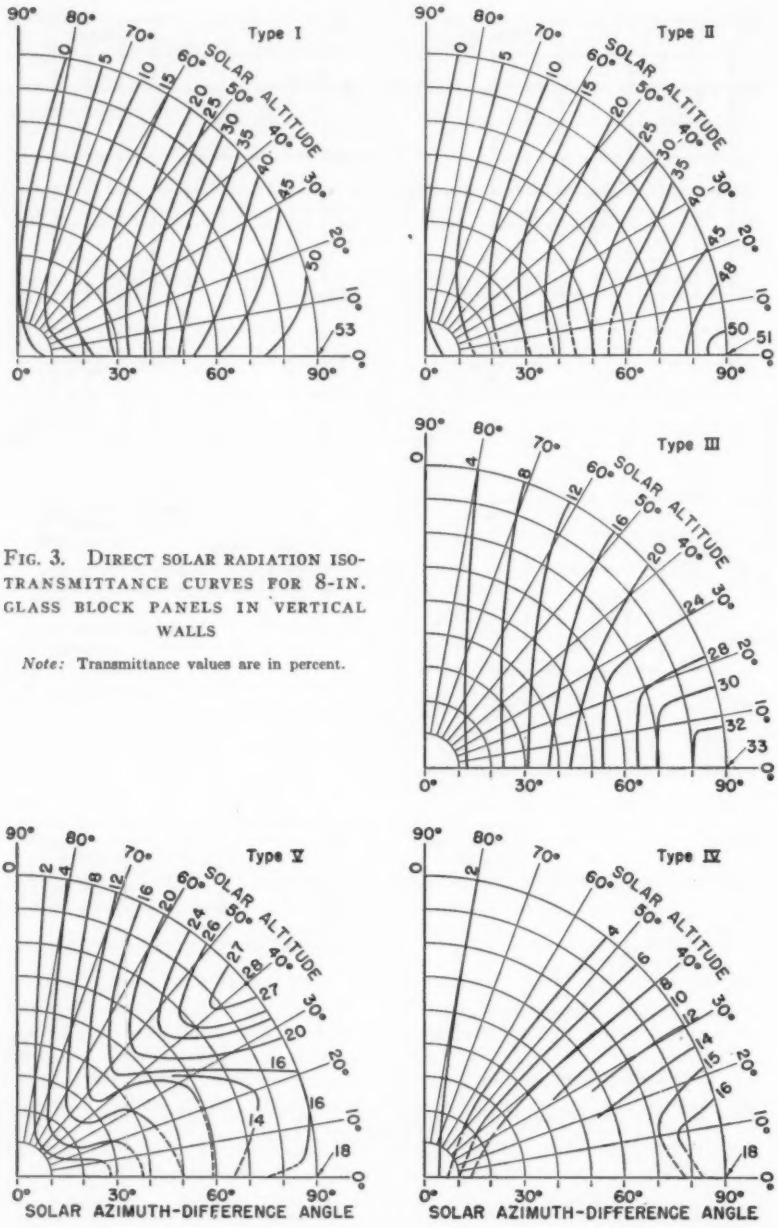


FIG. 3. DIRECT SOLAR RADIATION ISO-TRANSMITTANCE CURVES FOR 8-IN. GLASS BLOCK PANELS IN VERTICAL WALLS

Note: Transmittance values are in percent.

Because pattern effects are combined with mortar joint shading, the area within each pair of curves does not necessarily represent all the transmittances possible for any one angle of incidence. To include all possibilities, test data were plotted on polar coordinates with the radial scale as azimuth-difference angle and the angular position as solar altitude. The azimuth-difference angle is the angle between the projections of a wall and of the sun's rays on a horizontal plane. The solar altitude is the angle between the sun's rays and a horizontal plane. The values from the fixed angle tests were plotted together with test results from fixed position tests with south, southwest, and west orientations. These data gave a sufficient number of points, so that lines and areas of constant

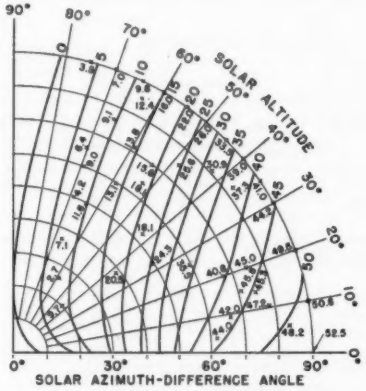


FIG. 4. COMPARISON OF TEST DATA WITH ISO-TRANSMITTANCE CURVES FOR 8-IN. TYPE I BLOCK

transmittance could be drawn. Fig. 3 shows the smoothed iso-transmittance curves for the five panels. Fig. 4 illustrates the agreement between the curves for Type I and the test points.

TABLE 3. TRANSMITTANCE OF DIFFUSE SOLAR RADIATION BY EIGHT-INCH GLASS BLOCK PANELS

(Measured by calorimeter)

PANEL TYPE	TRANSMITTANCE	
	Range	Average
I	0.22-0.33	0.27
II	0.18-0.32	0.27
III	0.11-0.26	0.19
IV	0.09-0.22	0.15
V	0.13-0.21	0.16

Heat Gains From Absorbed Solar Energy

A considerable portion of the incident solar radiation may be absorbed by a glass block panel, resulting in a rise in temperature of the outdoor and indoor surfaces. Part of this absorbed energy is stored in the panel, part is lost from the indoor surface by convection and radiation, and the rest is lost to the out-

TABLE 4. APPROXIMATE FRACTION OF INCIDENT SOLAR RADIATION WHICH ENTERS AN ENCLOSURE AT TIME OF MAXIMUM RATE OF HEAT ENTRY THROUGH 8-INCH GLASS BLOCK

(Zero Indoor-outdoor temperature differential; August 1; 40 deg north latitude)

Block Type	Orientation	Direct Incident Radiation				Diffuse Radiation		
		Approx. Hour (Sun Time)	Fractional Gain			Fractional Gain		
			Conv. & Rad. (Approx.)	Trans. (Measured)	Total (Approx.)	Conv. & Rad. (Approx.)	Trans. (Measured)	Total (Approx.)
I	South	1 p.m.	0.23	0.10	0.33	0.10	0.27	0.37
	West	5 p.m.	0.18	0.45	0.63			
II	South	1 p.m.	0.23	0.10	0.33	0.10	0.27	0.37
	West	5 p.m.	0.18	0.42	0.60			
III	South	1 p.m.	0.20	0.08	0.28	0.10	0.19	0.29
	West	5 p.m.	0.21	0.27	0.48			
IV	South	1 p.m.	0.17	0.03	0.20	0.15	0.15	0.30
	West	5 p.m.	0.19	0.15	0.34			
V	South	1 p.m.	0.17	0.14	0.31	0.15	0.16	0.31
	West	5 p.m.	0.18	0.17	0.35			

door surroundings by radiation and convection. The fraction absorbed depends upon:

1. The area and absorptivity of exposed mortar joints,
2. The angle of incidence,
3. The path length of the reflected and transmitted rays, and
4. The absorption coefficient of the glass.

The mortar joints, with the adjacent glass webs, form a path of relatively high thermal conductivity to the indoor side. The angle of incidence determines to a considerable extent the amount of radiation which passes through the outdoor surface, only to be trapped in the interior by the mortar joints. The fraction of the absorbed energy which leaves the indoor surface depends upon the indoor and outdoor boundary conditions for radiative and convective heat transfer and the heat capacity of the panel. The latter tends to diminish the amplitude of the heat wave and causes it to lag behind the time occurrence of the external effect which produces it.

In order to determine the diurnal cycle of solar heat gain, a number of tests have been made in which a panel was held in a fixed position. Most of these

tests were started before sunrise and were continued until after sunset. Data were taken at 15-min intervals.

It is not possible at this time to present an analysis of the heat gain due to solar energy absorption and its relationship to the factors previously mentioned. Instead, approximate values are given of the fraction of the incident solar energy that is dissipated from the indoor surface at the time of maximum total heat gain. This fraction has been determined by treating the data obtained in the fixed position tests in the following manner. The heat gain by the calorimeter via radiation and convection from the indoor surface of the test panel was computed in Appendix A. This quantity was then adjusted to a zero indoor-outdoor temperature differential by multiplying the value of U for the panel (see the next section) by the differential existing under test conditions. This product was added or subtracted if the outdoor temperature was lower or higher respectively than the indoor temperature. A second adjustment was made for the effect of absorbed diffuse radiation on heat gain by convection and radiation. The corrected value was then divided by the incident direct radiation. These results and the approximate time the peak gain occurs during midsummer at about 40 deg north latitude are presented in Table 4. Also listed is the fraction of the incident diffuse solar radiation which passes indoors by way of absorption and conduction. These values have also been corrected to zero indoor-outdoor temperature differential. The fraction of direct radiation transmitted at the time of maximum heat gain and the fraction of diffuse radiation transmitted are also listed.

Overall Coefficients of Heat Transfer and Panel Conductances

Table 5 lists the results of tests made out-of-doors to determine overall coefficients and panel conductances. Following previous practice[3] the overall coefficient of heat transfer, U, is based upon the heat flow through the panel per square foot area divided by the temperature difference between the calorimeter *surface* and the outdoor air. The panel conductance, C, is the same heat flow quantity divided by the temperature difference between the indoor and outdoor surfaces. In the tests, weather conditions were stable enough so that the panel was in thermal equilibrium with its surroundings for a period of several hours. The tabular values for these tests are averages of several one hour periods during which readings, taken at fifteen minute periods, showed no significant change.

The heat flow quantities were determined by averaging the results of two computations. In one, the radiation exchange and convection heat transfer between the calorimeter surface and the indoor side of the test panel were computed. The second was determined from the heat supplied to the calorimeter by the fluid system minus the heat leakages. The calculations follow the general procedure outlined for computing solar energy transmittances as given in Appendix A. In addition, small corrections, based upon heat meter measurements, were made to account for heat leakage at the perimeter of the panel. The two values checked within 2 to 7 percent.

Solar Energy Transmittance

In order to understand the contributions of face patterns and mortar joints to the transmitting characteristics of a given glass block panel, it is of interest

TABLE 5. OVERALL COEFFICIENTS OF HEAT TRANSMISSION, U, AND CONDUCTANCES, C, OF GLASS BLOCK PANELS

DATE OF TEST	TIME E.S.T.	BLOCK		WIND		SKY CONDITION	TEMPERATURES (FAHRENHEIT)					HEAT FLOW Btu per (Hr) (Sq Ft)	C Btu per (Hr) (Sq Ft) (F Deg)	U Btu per (Hr) (Sq Ft) (F Deg)
							Calorimeter		Panel Surfaces		Outdoor Air			
		Type	Size In.	Velocity mph	Angle of Incidence Deg		Surface	Air	Indoor	Outdoor				
4-5-49	1:00–5:15 a.m.	I	6	3.0	45	Clear	74.2	73.0	58.6	43.7	39.1	17.6	1.18	0.50
3-7, 8-49	10:45 p.m.–3:30 a.m.	I	8	3.5	0	Clear	72.3	70.9	56.0	37.7	31.9	18.0	0.98	0.45
3-16-49	7:00–8:15 p.m.	III	8	8.5	Sheltered	O'cast	74.1	72.9	59.0	35.7	29.3	17.2	0.74	0.38
3-23, 24-49	11:00 p.m.–2:30 a.m.	III	8	3.1	0	O'cast	75.5	74.5	62.4	44.6	37.0	14.5	0.81	0.38
2-28 to 3-1-49	10:45 p.m.–1:00 a.m.	V	8	7.8	90	O'cast	71.6	69.7	50.3	27.1	19.2	25.4	1.09	0.48

to compare test results with computed results. Consider the transmittance of an eight inch block consisting of two ¼-in. thick faces, 7¾ in. sq, flat and parallel and connected by 3¾ in. webs ½ in. wide. Such a block corresponds roughly to Type I, which has a fairly simple pattern on the cavity surfaces only. The average thickness of the block faces is approximately ¼ in. Table 2 shows that, at normal incidence, glass of this thickness transmits about 88 percent of

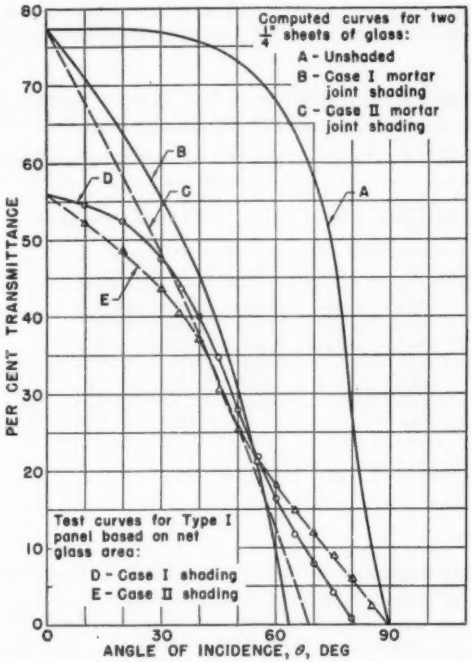


FIG. 5. COMPUTED TRANSMITTANCE FOR 8-IN. BLOCK COMPARED WITH TRANSMITTANCES OF TYPE I PANEL

normally incident radiation. The corresponding transmittance of the two faces is 77.5 percent; that of the web is 46 percent. The block as a whole would transmit about 66 percent. The corresponding transmittance of the Type I panel is 52.5 percent. A small part of difference is due to the fact that the web area, as measured at the outer surfaces, is not fully effective as transmitting area. The remainder of the difference probably can be attributed to the face pattern itself. Inspection shows that there are small areas of total or partial reflection distributed over the surface.

The shading effect of the mortar joints will now be discussed. To simplify the problem, the panels will be considered as made up of 7¾ in. squares of flat glass 3¾ in. apart with the glass web and mortar joint partitions replaced by thin non-reflecting partitions. Fig. 5 shows the computed transmittances for such a panel. Curve A is the transmittance curve for two sheets of ¼ in. glass. Curve B shows the shading effect of a single joint (Case I) and curve C shows the shading effect of two mortar joints (Case II). Curves D and E are the Case I and Case II curves of Fig. 2 for the Type I block adjusted for the mortar joint area. At angles of incidence in excess of 30 deg, the agreement is reasonably good in spite of pattern effects. This agreement can be explained by two

TABLE 6. COSINE OF INCIDENT ANGLE FOR VARIOUSLY ORIENTED VERTICAL WALLS AND A HORIZONTAL PLANE AND SOLAR ALTITUDE

(For 18 deg declination north (Aug. 1) and 40 deg north latitude)

LOCAL MEAN SUN TIME		SOLAR ALTITUDE, DEG	ORIENTATION OF WALL						
A.M. ➔ ▼			N	NE	E	SE	S	SW	Horiz.
5	7	0.5	0.406	0.934	0.914	0.358	.....	.....	0.010
6	6	11.5	0.237	0.840	0.951	0.505	.....	.....	0.199
7	5	23.0	0.079	0.705	0.919	0.594	.....	.....	0.388
8	4	34.5	.....	0.533	0.824	0.631	0.069	.....	0.564
9	3	45.5	.....	0.337	0.673	0.614	0.196	.....	0.713
10	2	56.0	.....	0.129	0.475	0.542	0.292	.....	0.830
11	1	64.5	.....	.....	0.246	0.424	0.354	0.076	0.903
12		68.0	.....	.....	0.000	0.265	0.375	0.265	0.927
	▲ P.M. ➔		N	NW	W	SW	S	SE	Horiz.

facts: (1) if there had been a web in the ideal block, about half of it would have been eliminated as transmitting area by the shading effect of the joint, (2) the cavity surfaces are actually fair reflectors so that the rear or indoor surface of the block receives and transmits radiation that otherwise would never have reached it. This probably accounts for the fact that zero transmittance occurs at higher incident angles than would be predicted from the geometry of the block.

The curves for the Type IV and Type V block are extreme examples of the effect of face pattern. This is especially true of the Type V block, which has horizontal light directing prisms on the cavity faces. A drawing board analysis was made for incident angles of zero and 45 deg. The problem of tracing the rays through the block was simplified by considering the outer surface to be flat. It was found that at normal incidence, roughly 75 percent of the incident radiation was totally reflected. The reflected rays left the front face at an angle of about 20 deg, so that one would expect the transmittance at 20 degrees incident to be about the same as at zero incidence. The Case I curves are in agreement with this analysis. At 45 deg, there was little reflection except that normally associated with the various glass-air boundaries. The rays were bent upward at an angle, therefore tending to counteract the shading effect of the mortar

joint. The result was that, at 45 deg, the transmittance was about the same as for the Type I block.

The curves for the Type IV block show that the face pattern, designed to diffuse the incident radiation, reduces the transmittance to very low levels. The similarity of the Case II curve of this block and that of the Type V block is striking, but not at all unexpected. In both cases, the pattern presented to the sun's rays was essentially the same. Test data for the Type IV panel in the region of low solar altitude and the middle values of azimuth-difference angle were not extensive enough to permit extrapolation. Information recently received from the field indicates that the Type IV panel is often installed rotated

TABLE 7. AZIMUTH-DIFFERENCE ANGLE, DEGREES, FOR VARIOUSLY ORIENTED VERTICAL WALLS AND SOLAR ALTITUDE

(For 18 deg declination north (Aug. 1) and 40 deg north latitude)

LOCAL MEAN SUN TIME		SOLAR ALTITUDE, DEG	ORIENTATION OF WALL					
A.M.→ ↓			N	NE	E	SE	S	SW
5	7	0.5	24	69	66	21	.....	.....
6	6	11.5	14	59	76	31	.....	.....
7	5	23.0	5	50	85	40	.....	.....
8	4	34.5	.....	40	85	50	5	.....
9	3	45.5	.....	27	74	61	16	.....
10	2	56.0	.....	14	59	76	31	.....
11	1	64.5	.....	.....	35	80	55	10
12		68.0	.....	.....	0	45	90	45
	↑ P.M.→		N	NW	W	SW	S	SE

90 deg from the position in which the panel was set in the calorimeter. Until additional data are secured, the *curves as presented should be used only for installations corresponding to the test conditions.*

The pattern of Type II affects the transmittances about as would be expected and shows that its diffusing nature reduces the transmittance little as compared to Type I. The addition of a glass fiber screen, on the other hand (Type III), markedly reduces the transmittance.

Use of the Transmittance Curves

Fig. 3 makes it possible to select the transmittance of vertical panels for any orientation, location, and time of day. With suitable data relating direct solar intensity to solar altitude, the amount of solar radiation transmitted can be determined satisfactorily. In view of the data given in Table 2, it is believed that the effect of variations in the distribution of solar energy on transmittance is not large.

When solar intensity data can be correlated with other weather design data, design tables of instantaneous rates of heat gain can be made up for each block

type for various localities throughout the country. As an interim measure and one offering some degree of flexibility, it is suggested that Fig. 3 be used with tables such as Tables 6 and 7. Table 6 gives the solar altitudes and cosines of the incident angle for variously oriented vertical walls at 40 degrees north latitude with a solar declination of 18 degrees north (Aug. 1). Table 7 gives

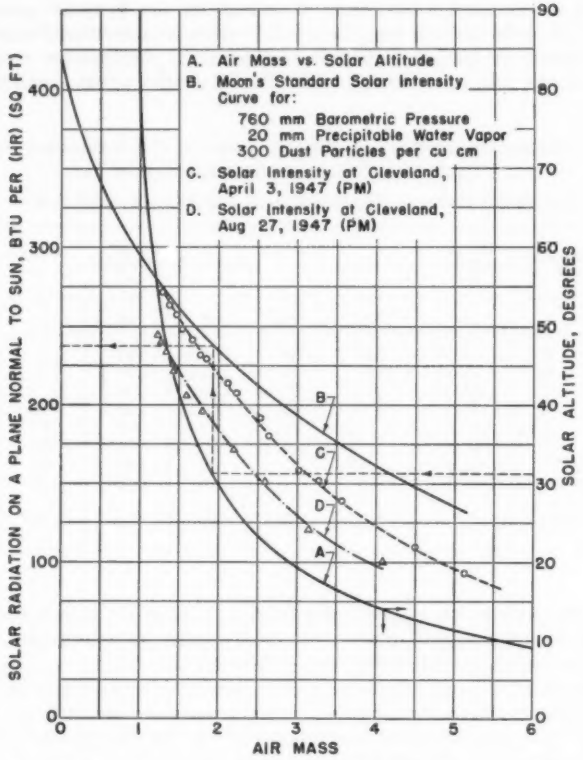


FIG. 6. INTENSITY OF DIRECT SOLAR RADIATION ON CLEAR DAYS AT CLEVELAND

solar altitudes and the azimuth-difference angles for the same latitude and season. Use of these tables for any time between May 1 and August 15 will not lead to serious error. These tables are similar to Tables 8 and 22 of Chapter 15 of the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1949.

Fig. 6 presents several curves (B, C and D) of direct solar radiation intensity on a plane normal to the sun's rays, plotted against air mass which, at sea level and standard pressure, correspond to the solar altitudes given on the right hand

ordinate. Air mass at any solar altitude may be found from curve A Fig. 6. For elevations above sea level, the air mass corresponding to a given solar altitude is reduced in direct proportion to the barometric pressure for the given station. Values given by curve B in most cases are identical with those in Table 5 Chapter 15 of THE GUIDE 1949 and are the basis of the present tables of design solar heat gain for single ordinary window glass. The curves for Cleveland illustrate to some extent the variations in intensity possible on cloudless days. With data such as these and Table 6, the irradiation of a wall can be readily computed.

The diffuse intensity on a vertical wall used in Table 18 of THE GUIDE is of the order of 20 Btu per (hr) (sq ft). It is of interest to note that similar values were observed in Cleveland in 1948 when the solar intensity was somewhat in excess of curve B. On the other hand, the diffuse values associated with curve D were of the order of 40 Btu per (hr) (sq ft). An example illustrating the use of Fig. 3 and Tables 6 and 7 may now be given.

Find the solar energy transmitted by a wall of Type I block facing southwest at 2 p.m. sun time.

From Table 6 the cosine of the incident angle is 0.542; the solar altitude is 56 deg. From Fig. 6 the standard direct solar intensity on a normal plane is 280 Btu per (hr) (sq ft); on the vertical plane it is 0.542 times 280 or 151.7 Btu per (hr) (sq ft). From Table 7 the azimuth-difference angle is 76 deg. For this angle and an altitude of 56 deg, the transmittance is 0.18. From Table 3 the diffuse transmittance for this type of block is 0.27. The total solar energy transmitted is 0.27 times 20 plus 0.18 times 151.7, or a total of 32.7 Btu per (hr) (sq ft). It should be realized that this instantaneous gain does not necessarily produce an equal instantaneous load on the system because of internal heat storage effects.

Total Heat Gain

It is emphasized that the values of solar energy gain by convection and radiation from the indoor surface as given in Table 4 are rought approximations only. Although most of the values were determined directly from test conditions, the boundary conditions governing heat transfer by radiation and convection and the cyclic variation in the solar intensity were by no means identical.

Therefore, the values for one type of block do not necessarily indicate its performance relative to another type or even the same block with another orientation. Further analysis of the data is required to determine with more precision the relationship of heat gain to boundary conditions.

As pointed out in previous papers and elsewhere in the literature, the heat gain data determined by these tests are instantaneous rates of heat release to a conditioned enclosure. Because of the quick response of the apparatus, they were also instantaneous cooling loads on the calorimeter system. This is far from true in actual installations. Because the interior structure stores up much of the solar energy directly transmitted by a glass block panel and that emitted by the panel as low temperature radiation, the load imposed on a cooling system is ordinarily much less than the instantaneous heat gain from the panel. The effect varies with the distribution of radiant energy transmitted by the panel, the heat capacity of the structure. and the method of cooling.

Overall Coefficients of Heat Transmission

Because overall coefficients of heat transmission are dependent upon the indoor and outdoor boundary conditions for convective and radiative heat transfer, the fundamental heat conducting characteristic of a given glass block panel is best indicated by its conductance. Inspection of the data of Table 5 shows that the conductances of eight inch block panels, with the exception of Type III, average about 1.1 Btu per (hr) (sq ft) (F deg). Since wind velocity and direction and radiation exchange affect the surface temperature distribution, the panel conductances may be dependent to some extent on these influences. This is indicated by the values given for the Type III block and shows that additional tests would be desirable.

The conductance of the Type III panel is markedly lower because of the glass fiber screen which divides the cavity. This screen principally affects the transfer of radiant energy across the cavity. Since the air pressure in the cavity is approximately one-third of an atmosphere, the convection heat transfer across the cavity is probably so small that the addition of the screen can have but little effect.

SUMMARY

Transmittances of five eight-inch glass block panels have been determined for direct and diffuse solar radiation by calorimeter tests. Transmittance curves and suggested tables of solar angles for design use have been presented. Estimates of the amount and time of maximum solar heat gain for south and west walls have been given. Overall coefficients of heat transmission have been determined under steady state, natural weather conditions by means of the calorimeter apparatus. The results of the investigations may be summarized as follows:

1. The shading effect of the mortar joints is a principal factor in establishing the solar energy transmitting characteristics of glass block panels, except in the case of prism-type blocks. Increasing obscurity of pattern design and use of screens reduce energy transmittance proportionately.

2. The light directing characteristics of the prism-type blocks cause their transmitting characteristics to depart considerably from the trend indicated by mortar joint shading.

3. Heat gain data due to absorbed solar energy must be recognized as approximations that are subject to considerable variation and therefore must be used with caution.

4. If design boundary conditions of heat transfer had existed during the tests, values of overall coefficients for glass block panels would be somewhat higher than those presented. The possible influence of boundary conditions on panel conductance needs further study.

ACKNOWLEDGMENT

The authors acknowledge with thanks the assistance and counsel given by their colleagues of the A.S.H.V.E. Research Laboratory staff, the contribution of the American Structural Products Co. and the Pittsburgh-Corning Corp. in providing technical data and assistance, and the help of the members of the A.S.H.V.E. Technical Advisory Committee on Heat Flow through Glass for

stimulating interest in the project, and also providing both suggestions for procedure and technical information.

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APPENDIX A

SAMPLE DATA AND CALCULATIONS TO DETERMINE TRANSMITTANCE OF DIRECT SOLAR RADIATION

(Observed Data for Nov. 8, 1948, 11:00-11:30 a.m. EST)

Temperatures (converted from observed thermocouple emfs); F deg:

Calorimeter Air (at 18 in.)	70.9
Calorimeter Surface, Avg.	68.5
Calorimeter Surface, Start of Run	68.5
Calorimeter Surface, End of Run	68.7
Glass Block, Inside	82.1
Mortar, Inside	83.3
Average Panel Inside	82.2
Mean Panel and Calorimeter Surface	75.4
Glass Block, Outside	75.6
Mortar, Outside	83.0
Average Panel Outside	76.1
Outdoor D.B. (couple)	56.3
Outdoor D.B. (sling)	55.0
Outdoor W.B. (sling)	44.0
Casing of Calorimeter	57.5
Coolant, mean	65.4
Test House Roof (in sun)	80.4

Other Observations:

Emf of coolant thermopile	1.409 mv
Emf across calorimeter insulation	1.471 mv
Emf of calorimeter frame heat meters	0.207 mv
Emf drop across calorimeter flange	0.047 mv
Emf of panel frame heat meters	0.487 mv
Incident angle	20 deg
Angle of tilt	15.8 deg
Sunsight angle (Case II)	135 deg

Azimuth angle	0 deg (south)
Pyrheliometer A, total radiation	3.137 mv
Pyrheliometer A, diffuse radiation	0.398 mv
Pyrheliometer B, diffuse radiation	0.977 mv
Coolant flow, Hg, large orifice	9.63 in.
Wind velocity	4.3 mph
Wind direction	S.E.
Cloudiness, tenths	5

Total Heat Gain by Calorimeter [all values Btu per (hour) (square foot)]:

Coolant Flow Rate = Chart value × density correction factor × temperature correction factor
= 860 × 1.002 × 0.998 = 860 lb/hr

Coolant Heat Gain = (lb/hr × specific heat × thermopile emf/24 × deg/mv) ÷ 16.0
= (860 × 0.776 × 1.409/24 × 45.9) ÷ 16.0 = 112.2

Heat Leakage Corrections:

Insulation = (Area × k/L × emf/13 × deg/mv) ÷ 16.0
= (40.6 × 0.023/0.167 × 1.471/13 × 45.9) ÷ 16.0 = 1.80

Frame = (Area × emf × temperature factor × meter constant) ÷ 16.0
= (7.33 × 0.207 × 1.065 × 15.2) ÷ 16.0 = 1.54

Flange = (Area × k/L × emf/12 × deg/mv) ÷ 16.0
= (2.10 × 117/0.125 × 0.047/12 × 48.1) ÷ 16.0 = 1.90

Calorimeter Shell Storage = (Wgt, lb × specific heat × temperature rise/hr) ÷ 16.0
= (11.8 × (68.7 − 68.5)/30 × 60) ÷ 16.0 = 0.30

Unaccounted-for loss (by calibration) = function of calorimeter surface temperature-casing temperature for 68.5 − 57.5 = 11.0 differential = 0.65

Total Heat Leakage = 6.19

Corrected Total Heat Gain = Coolant Gain + Heat Leakage = 118.39

Solar Irradiation [all values Btu per (hour) (square foot)]:

Direct Radiation = ("A" total mv − "A" diffuse mv) × constant × factor for angle
= (3.137 − 0.398) × 81.3 × 0.987 = 220.0

Diffuse Radiation = ("A" mv diffuse × constant + "B" mv diffuse × constant) ÷ 2
= (0.398 × 82.5 + 0.977 × 32.8) ÷ 2 = 32.5

Heat Transfer Inside Panel Surface to Calorimeter [all values Btu per (hour) (square foot)]:

By radiation:

Radiation coefficient at 75.4 F mean temperature = 0.98 Btu/ (hr) (sq ft) (F deg)
Heat Flow by radiation = radiation coefficient × temperature difference = 0.98 (82.2 − 68.5) = 13.43

By convection:

Heat flow by convection = vertical surface value × factor for tilt
Vertical surface value = 3.0 for (82.2 − 68.5) temperature difference
For tilted surface** = 3.0 × 0.92 = 2.76
Total flow from panel by radiation and convection = 13.43 + 2.76 = 16.19

** See Fig. 3 in Reference 3.

Solar Energy Transmitted [*all values Btu per (hour) (square foot)*]:

Total transmitted	= total gain by calorimeter − heat flow from panel = 118.39 − 16.19 = 102.20
Diffuse transmitted	= incident diffuse × transmittance separately determined = 32.5 × 0.27 = 8.75
Direct transmitted	= total transmitted − diffuse transmitted = 102.20 − 8.75 = 93.45
Percent direct transmittance	= direct energy transmitted ÷ incident direct radiation = 93.45 ÷ 220.0 = 42.5 percent

DISCUSSION

A. H. BAKER*, Port Allegany, Pa. (WRITTEN): Spectral transmittance curves were obtained for the various glasses used in the tests described in this paper. The general purpose was to determine what effect the spectral transmittance of the glass itself has upon the overall solar energy transmittance of the glass block wall construction.

Therefore, I would suggest that the spectral transmittance curves be presented and that the authors discuss their significance.

E. W. CONOVER, Detroit, Mich. (WRITTEN): The authors of this paper have made a valuable contribution to the available information on glass block panels and it is hoped that the data can be worked up into simplified forms for design purposes. We will leave the discussion of the technical parts of this paper to those more familiar with the transmission characteristics of glass and confine ourselves to points not clear to us.

The paper states that Table 1 is a description of the panels. This describes the individual blocks rather than the panels tested.

Tables 2 and 3 give transmittances for various glass samples with no unit specified. Is this the percentage of the normally incident solar radiation?

The description of the azimuth-difference angle mentions the projections of a wall. This seems ambiguous and needs clarification, preferably by a sketch.

We question the accuracy of the method of computing the approximate fraction of incident solar radiation values in Table 4 when using the U value obtained during night tests for the panels heated up by solar radiation. A different U value would appear logical for this condition.

In explanation of the agreements between computed transmittances for a hypothetical block panel and tests, as shown in Fig. 5, the two facts given appear to explain the disagreements instead of the agreements.

We would like to ask what is *the irradiation of a wall.* We assume that the Table 18 of THE GUIDE referred to in connection with diffuse intensity is from Chapter 15, but we can find no reference therein giving the 20 Btu as a basis for this table.

Table 5 gives overall coefficients of heat transmission U for the glass block panels as determined by night tests. These are all for moderately light wind velocities and no attempt is made to correct them for design conditions of a 15-mile wind outside and natural convection on the room side. Since these test figures are very close to those now given in THE GUIDE, it appears that THE GUIDE figures are now too low. Since it is stated that the panel conductances were influenced by the wind velocity and direction and the radiation exchange, it seems desirable to make additional tests

* Technical Director, Pittsburgh Corning Corp.

to determine these and be able to submit adequate and acceptable data for revision of THE GUIDE.

In discussing the lower conductances of the Type III panel, it is stated that the air pressure in the cavity is approximately one-third of an atmosphere. How was this obtained and is it considered as representative of all glass block?

D. D'EUSTACHIO†, Port Allegany, Pa. (WRITTEN): This paper is important because of the great care that has been taken to get reliable data. Hence, the results should be reduced to a form applicable to practice as rapidly as possible. One step in this process will require defining the amount of variation in block patterns, from that given in Table 1, which can be permitted without changing performance. As an example, Type IV block is frequently installed with corrugations in a vertical direction instead of horizontal. How could your results be applied to this situation? Along this same line, have you given any thought to developing a single test that could be used to classify new block designs into your present grouping?

J. N. LIVERMORE, Detroit, Mich. (WRITTEN): I wish to comment briefly on this paper as one who spends most of his working hours in a building whose walls are made almost entirely of glass block. Therefore, in a sense, glass block is a part of my daily existence. Moreover, I was a party, 10 years ago, to the design of the air conditioning system which was installed in this same building. Recalling the sketchy state of glass block heat transmittance data at that time brings only regret that the authors of this paper had not gotten around to this work long ago. It has been badly needed.

The authors hint that there is further work to be done on this project. Before completing it, I have two suggestions to make:

1. That they make an effort to separate the radiant and convected heat release from the interior surfaces of the glass block when exposed to the sun. I believe Table 4 shows the two combined. This separation may prove of considerable value in studies now contemplated on the relation of interior radiant heat gains to cooling apparatus load.
2. I suggest that this paper should include a practical example in which the authors show the method that they expect engineers to follow in applying these data to an actual problem.

J. P. STEWART, Syracuse, N. Y. (WRITTEN): The authors have added another valuable chapter to solar transmission through glass materials. It is very encouraging to note that transmittances measured by pyrheliometer agree so well with computed values from spectro-photometric data. I would like to see these most useful data, heat gain through glass block, arranged in some simplified tabular form for quick reference in THE GUIDE 1949. It seems proper to show basic tables giving the instantaneous rates of heat release to the air conditioned space. Modifying factors could be given as judgment values in estimating the actual cooling load on a particular installation, taking heat storage into consideration.

Could the authors give the approximate time lag in the convection and radiation heat flow through the glass block for heat gain due to direct incident radiation? The two typical curves C and D for Cleveland, Fig. 6, are both below curve B which represents Moon's standard solar intensity for sea level. Does this indicate that the standard design curve for solar radiation used in the A.S.H.V.E. GUIDE 1949 is, in general, too great for a clear, humid day in summer, at least for Cleveland.

H. B. VINCENT, Toledo, Ohio (WRITTEN): The progress which has been made on a fundamental research project is often not too apparent to those not immediately

† Physicist, Pittsburgh Corning Corp.

associated with the work. The ingenuity shown in improvements in instrumentation and measurement procedure has been touched upon by the authors very modestly.

It would appear worthwhile to mention that the transmittance measurements reported were made with a new type of equipment developed by the authors to have a very short time constant so that instantaneous values of heat input are indicated. Results should be reproducible to a much higher degree than in previous measurement made with equipment having a large thermal capacity and involving a substantial time lag which is difficult to estimate.

All instruments used in the work have been investigated exhaustively and many have been redesigned to increase reliability. In this connection, special mention may be made of improvements in shielding of radiometers to reduce the effects of convection.

In the selection of block for heat transmittance measurements, an effort to choose samples which would be representative of designs currently manufactured has led to work on four types described in the paper. Any other design may be classified as similar to one of the four measured types and assumed to have an equal transmittance. Such an assumption will involve certain errors, but these may be small enough to make the procedure acceptable. Comments by the authors on this point would be appreciated.

Work is in process looking toward the application of the measurements reported to typical room structures. In the meantime, care should be exercised in any attempt to draw conclusions relative to operation in buildings. It must be recognized that the measurement device differs fundamentally from a structure designed for occupancy so that surrounding temperatures, convection effects, input to a cooling system and the like may be expected to calculate to values not readily recognizable from instantaneous heat flow data.

It is a pleasure to compliment the authors on the progress shown in their work and to wish them and the laboratory a successful continuation.

AUTHORS' CLOSURE: In reply to Dr. Baker, the authors submit Fig. A, spectral transmittance curves of glass samples, one-half inch thick, made from the melts used in the manufacture of the blocks studied at the Laboratory. The normal incidence transmittances are seen to be relatively constant over that part of the solar spectrum which contains most of the sun's energy, namely, 0.4 to 1.0 microns. Therefore, one would expect that differences in the energy distribution of the incident radiation would affect the transmittance values no more than the probable percentage of error in the determinations with the calorimeter apparatus. This percentage of error is estimated to be of the order of 3 to 5 percent, and is dependent somewhat upon test conditions. Possible exceptions to this statement are the Type IV and Type V blocks where the transmittance is reduced by the reflection characteristics of the cavity prisms. At some angles, the path length becomes long enough due to internal reflections that the absorption is high and transmittance much lower in the infrared than in the visible portion of the spectrum.

Mr. Conover quite properly questions the correctness of adjusting to a zero indoor-outdoor temperature differential the gain by convection and re-radiation from sunlit panels by using U values from night tests. Our data on outdoor surface conductances are not yet accurate enough that much refinement is justified at this time. In making the adjustment, no attempt was made to allow for differences in wind velocity and radiant exchange conditions. In all cases, the adjustment was of the order of 5 to 10 Btu per (hr) (sq ft) except for the panel of Type III block. The fractional gain values for this block must be regarded as being subject to considerable possible error because 30 to 50 deg temperature difference existed in these tests. Though expressing radiant and convected heat gain data as fractional values of the incident radiation is a convenient method when one is concerned with peak heat gain, it has little value for

later hours of the day because of time lag effects. After sundown this fraction can become infinite.

Mr. Conover states that the data on overall coefficients indicate that values of U in THE GUIDE may be too low. Since the panel conductances are low, a considerable increase in surface conductances will cause a relatively smaller increase in the U value. There is still some question as to how much a given change in surface conductance will affect panel conductance. More data, therefore, must be obtained to determine what changes can be made in values shown in THE GUIDE.

The approximate pressure of the block cavity was obtained from the manufacturers and presumably was estimated from a knowledge of the temperature of the block at the time the two halves are pressed together for sealing.

Whether the explanation of curves of Fig. 5 explain disagreement instead of agreement is probably a matter of viewpoint. A better understanding of mortar joint

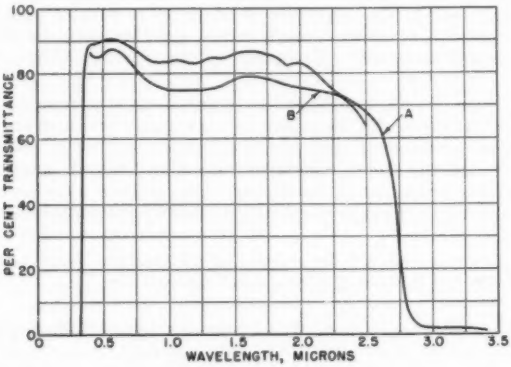

FIG. A. SPECTRAL TRANSMITTANCE CURVES OF GLASS SAMPLES

shading and pattern effects would have been obtained if a clear block had been studied. Perhaps this can be done as a part of future work.

There is no direct reference in THE GUIDE to the value of 20 Btu per (hr) (sq ft) diffuse irradiation of vertical surfaces used in making up the data for single window glass in THE GUIDE, Chapter 15, Table 18. This is approximately the value agreed on by C. M. Ashley and the senior author when the former prepared a revision of Chapter 15 some years ago and the latter prepared Table 18. Actually, the value was made to vary with solar altitude.

In reply to Dr. D'Eustachio's comments regarding the transmitting characteristics of the Type IV block when the corrugations are vertical, the authors submit Fig. B. This shows iso-transmittance curves for this type block with the corrugations vertical; that is, prisms horizontal. Additional test work completed after the paper was submitted for publication made it possible to give more complete data on this block than were given in the paper. Considerable computation is required to transpose curves obtained for a given block in one position to curves that show its performance in a position which is 90 deg from the first.

The authors can suggest no wholly satisfactory single test that could be used to classify new block designs into the present grouping of five types. However, since

light is in some ways easier to measure, it might be possible to insert a single block in the opening of an intergrating chamber surfaced with a non-reflective light sensitive material. If this surface were calibrated in terms of the light source, transmittance might be determined for various angles. The convection and radiation gain could be closely estimated from the data for the block which applies most closely to the unknown.

Mr. Livermore suggests that the radiant heat and the convected heat released from the interior surfaces of the sunlit glass block be separated to facilitate relating instantaneous loads to apparatus load. This has been done in working up the test data for this report and all previous reports. A sample set of data given in the Appendix

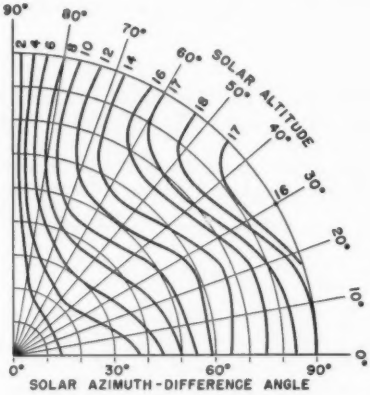

FIG. B. ISO-TRANSMITTANCE CURVES FOR TYPE IV BLOCK WITH VERTICAL CORRUGATIONS

shows these computations. In the calorimeter apparatus used, the convected and radiated heat are roughly in the ratio of one to five. Convection is low because air motion is due entirely to natural means. In actual buildings, the air motion due to air distribution systems would cause this ratio to be more nearly one to two.

The authors would like to have elaborated on the results given in the paper and to have shown how the data might be applied to actual engineering problems. An attempt was made to illustrate the use of the iso-transmittance curves, the solar intensity curves, and the tabular values of altitude and azimuth-difference angle. However, lack of space and the necessity of meeting a deadline made it impossible to do more than this.

In reply to Mr. Stewart, the convection and radiation heat flow lagged by about one hour behind the incident radiation curve for south and west walls. The fact that curves C and D of Fig. 6 are both below Moon's Standard Solar Intensity curve does indicate that, for the Cleveland area, Moon's design curve, which is used in the 1949 GUIDE, is too high for days which produce the greatest cooling load. On the other hand, for many parts of the country, Moon's suggested intensity values are too low.

Dr. Vincent's question with regard to classifying other designs into the categories

set up in the paper is in the same vein as Dr. D'Eustachio's. For those patterns currently manufactured, it would seem that most would fall into Type I or Type II. For example, those with a simple pattern or no pattern at all could be classed Type I with the transmittances between 0 and 25 deg incident angle adjusted upward a bit in the case of block with no pattern.

The patterns with prisms are much more difficult to classify, since transmittance appears to be so dependent upon prism angle and orientation. For example, the Type IV block with prisms horizontal, for all practical purposes, is an entirely different block than the same block with the prisms vertical.

Dr. Vincent quite properly points out that the input to the cooling system is not readily recognizable from the instantaneous heat flow data. Professors Mackey and Gay, in their paper, Heat Gains are Not Cooling Loads, (see page 413), enable one to make some estimate of the influence of internal storage effects on heat gain of a radiant nature.

1375

No. 1375

STANDARDIZATION OF THERMODYNAMIC PROPERTIES OF MOIST AIR†

Final Report of Working Subcommittee, International Joint Committee on Psychrometric Data

By JOHN A. GOFF*, *Chairman*

INTRODUCTION

THE inaugural meeting of the International Joint Committee on Psychrometric Data (I.J.C.P.D.) was held in Boston, Massachusetts, on January 23, 1945, in connection with the 51st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. In November, 1945, the late J. Herbert Walker, temporary chairman, appointed a Working Subcommittee, of which the writer was privileged to be Chairman, to assist in the drafting of an agenda for the next meeting of his Committee.

The Working Subcommittee met first in Philadelphia on February 16, 1946. The minutes of this first meeting were used as an agenda for the second meeting held in Washington, D. C., on November 26, 1946, and attended by a number of interested non-members including representatives of the British Meteorological Office, the British Commonwealth Scientific Office, and the National Bureau of Standards. The minutes of this second meeting contain an item instructing the chairman to prepare a report of the Working Subcommittee's deliberations.

A preliminary draft of the above-mentioned report was submitted under date of April 14, 1947, to Mr. Walker who offered several valuable criticisms and suggestions for incorporation in the final draft. Not until now, however, has the writer found it possible to prepare this final draft. Meantime the preliminary draft has served a very useful purpose as will be noted.

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† This report was accepted by the International Joint Committee on Psychrometric Data at its meeting on May 12, 1949. Chairman of the Joint Committee is Dr. Baldwin M. Woods, past president of the A.S.H.V.E. and director of University Extension, University of California, Berkeley.

* Dean, Towne Scientific School, University of Pennsylvania. The other members of the Working Subcommittee are: C. S. Cragoe, National Bureau of Standards; L. P. Harrison, U. S. Weather Bureau; F. G. Keyes, Massachusetts Institute of Technology. S. Gratch, Towne Scientific School, University of Pennsylvania, rendered invaluable aid to the Working Subcommittee as technical adviser.

At the first full session of the Aerological Commission of the International Meteorological Organization (I.M.O.) which met in Toronto, Canada, during August, 1947, there was appointed a Subcommission I on Functions and Tables under the chairmanship of Prof. P. A. Sheppard, Department of Meteorology, Imperial College of Science and Technology, University of London. By its terms of reference this Subcommission was charged with the task of recommending a set of self-consistent, internationally acceptable values of physical functions and constants of common use in meteorological practice. It is exceedingly gratifying to note the extent to which those recommendations of the Subcommission having to do with the thermodynamic properties of moist air agree with the recommendations of the Working Subcommittee as set forth in the preliminary draft referred to in the preceding paragraph. The former were adopted officially by the Twelfth Conference of Directors of the International Meteorological Organization sitting in Washington, D. C., September, 1947.

The report of Professor Sheppard's Subcommission is soon to be printed and widely distributed by the International Meteorological Organization as Document No. 71 of its Aerological Commission. It seems especially appropriate, therefore, to publish the final report of the Working Subcommittee of the International Joint Committee on Psychrometric Data more or less simultaneously. There is ample justification for the *A.S.M.E.* Special Research Committee on Properties of Gases and Gas Mixtures to sponsor the presentation and publication of this final report in the fact that it deals with the thermodynamic properties of one of the most common and technically important gas mixtures, namely, moist air.

It is to be deeply regretted that this report must record the death late in 1947 of Mr. Walker, Temporary Chairman, International Joint Committee on Psychrometric Data. It was upon his recommendation that Council of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS invited *all organizations likely to be interested in the problem of the properties of mixtures of air and water vapor to appoint a representative on a joint Inter-Society Committee to consider the adoption, when available, of basic equations and tables of such properties.*[1]

PROBLEM

The problem before the International Joint Committee on Psychrometric Data may be regarded as consisting of three parts: (A) to obtain information regarding the thermodynamic properties of moist air which can claim general acceptance as standard on the ground that it is thermodynamically consistent within the accuracy of existing knowledge; (B) to develop specifications of standard instruments for the practical realization of these properties in the field; (C) to recommend standard symbols, terminology, and charts calculated to promote a better understanding of underlying theory in its application to practical problems involving moist air.

This report sets forth the recommendations of the Working Subcommittee on Part A of the overall problem and explains the reasoning on which these recommendations are based.

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[1] Exponent numerals refer to Bibliography.

CRITERION

The Working Subcommittee agreed from the start to try to recommend a formulation, rather than discrete tables, of the thermodynamic properties of moist air, valid over the widest practical ranges of pressure and temperature. While for the majority of present day applications in the field of air conditioning it might suffice to recommend a table of properties at standard atmospheric pressure, such a table would by no means satisfy the needs of the meteorologist or the aerodynamic engineer; moreover, it would be shortsighted to assume that, even in the field of air conditioning engineering, the number of applications in which accurate properties at other pressures are needed will not increase substantially as time goes on. Fortunately, the pressure range of principal interest now and probably for some time in the future is the low-pressure range where well-tested theory is able to make reliable predictions regarding the effect of pressure on the various thermodynamic properties of moist air.

The working Subcommittee found no difficulty in adopting *thermodynamic consistency within the accuracy of existing knowledge* as the principal criterion by which to judge the validity of its recommendations. The adoption of this criterion at once implies the abrogation of Dalton's Law on which most predictions of the thermodynamic properties of moist air have heretofore been based. By present day standards Dalton's Law must be regarded as an inaccurate conjecture based upon an unwarranted faith in the *ultimate simplicity of nature.* Granted that the law and the consequences inferred more or less intuitively from it are extremely useful as rough and sometimes even close approximations to the truth, their use as a basis for standardization is now completely out of the question.

DALTON'S LAW

Dalton's Law is a difficult thing to discuss in any clear and unequivocal manner because such a variety of interpretations have come to be placed upon it. If the statement of it given by Preston[2] in his classical treatise, *The Theory of Heat,* be accepted it would appear that the pressure is the only property of a gas mixture which the law attempts to predict; and this, it says, is the sum of the pressures which the constituents would individually exert if each occupied separately the volume of the mixture at the temperature of the mixture. Now statistical mechanics predicts that the law would hold identically if each constituent were a hypothetical perfect gas showing no effects of intermolecular forces whereupon (*a*) the individual pressures would be the individual molfractions times the pressure of the mixture, (*b*) the sum of the individual enthalpies would equal the enthalpy of the mixture, and (*c*) the sum of the individual entropies would equal the mixture entropy.

Unfortunately, the preceding predictions (*a*), (*b*), and (*c*), which are strictly valid only in the case of a hypothetical mixture of perfect gases, have been used as vehicles for the introduction of various concepts that cannot be applied without ambiguity or thermodynamic inconsistency to mixtures of actual imperfect gases. It may be stated that Dalton's Law regarding the additivity of individual pressures and its alleged consequences regarding the additivity of individual enthalpies and entropies fails mainly, if not entirely, because of the

effects of intermolecular forces between *unlike* molecules while the alleged consequences that the individual pressures are the individual mol-fractions times the pressure of the mixture fail because of the effects of forces between *like* molecules also.

Various modifications of Dalton's Law have been proposed from time to time including the so-called Gibbs-Dalton, Amagat, and Lewis-Randall rules. None of these, however, offers any substantial improvement over Dalton's Law itself; all of them must now be regarded as conjectures based pretty much on wishful thinking.

STATISTICAL MECHANICS

Developments in statistical mechanics have given us a well-tested theory of the thermodynamic properties of gas mixtures which is entirely workable in the range of sufficiently low pressures. This theory can be expressed succinctly in the following form,

$$\mu_i = g^o_i(T) + RT\log_e(x_i p) + p\,[\Sigma x_i x_k A_{ik}(T) - 2\Sigma x_k A_{ik}(T)] + \quad . \quad . \quad . \quad (1)$$

where: the $g^o_i(T)$ are pure temperature functions called *zero-pressure reduced free enthalpies* which are directly calculable from a knowledge of the spectroscopic constants of the atomic or molecular species in question; the x_i and x_k are mol-fractions; the $A_{ik}(T) = A_{ki}(T)$ are pure temperature functions called *second virial coefficients* $(i = k)$ or *interaction coefficients* $(i \neq k)$ in terms of which the effects of intermolecular forces are expressed. The *chemical potentials* μ_i, when expressed as functions of temperature, pressure, and the various mol-fractions, are characteristic functions in the sense that from a knowledge of them can be derived all thermodynamic properties of the homogeneous phase to which they belong by application of such identical relations as the following:

$$g = \Sigma x_i \mu_i; v = \partial g/\partial p; s = -\partial g/\partial T;$$
$$h = \partial(\tau g)/\partial \tau \quad . \quad . \quad .

Accordingly the total concentration of the residual part of atmospheric air should be expected not to exceed 0.0025 percent.

The gases in the first group comprise what is commonly called *dry air*. It is true that the relative composition of this group is subject to secular variations and variations with location, meteorological conditions, and season, but these variations are produced mainly, if not entirely, by variations in the concentration of carbon dioxide. All evidence indicates that the average concentration of carbon dioxide in dry air in the troposphere is very close to 0.03 percent. Carpenter[4] recommends the value 0.031 percent.

For the purpose of standardization the composition of dry air can be defined more or less arbitrarily. It would seem desirable, however, to base the definition on available information regarding the average composition of atmospheric air in the troposphere. Carpenter[4] finds that dry air contains 20.939 percent oxygen and 0.031 percent carbon dioxide. Paneth[3] accepts the value, 20.95 percent oxygen in dry air containing carbon dioxide, recommended by Benedict[5]. The National Bureau of Standards has come to prefer the values, 20.94 percent oxygen and 0.03 percent carbon dioxide, which are the values originally agreed upon by the Working Subcommittee. It later developed, however, that the International Meteorological Organization holds a strong preference for the Paneth[3] values quoted previously. Hence, and particularly in view of the fact that the definition of what is to be referred to as *dry air* must be decided upon more or less arbitrarily, the writer agreed on behalf of his Working Subcommittee to accept the values, 20.95 percent oxygen and 0.03 percent carbon dioxide.

Cady and Cady[6] find that dry air free of CO_2 contains 0.93 percent argon by volume. This figure would not have to be modified appreciably to apply to dry air containing 0.03 percent carbon dioxide; hence it was accepted by the Working Subcommittee.

The (revised) recommendation of the Working Subcommittee regarding the composition of dry air is as follows:

Oxygen (O_2)	0.2095
Nitrogen (N_2)	0.7809
Argon (A)	0.0093
Carbon Dioxide (CO_2)	0.0003
	1.0000

This is a *fixed, mol-fraction* composition which the Subcommittee intends should be regarded as *exact by definition*. The fact that the foregoing figures were derived from experimental determinations made with an Orsat-like apparatus which does not yield information regarding the mol-fractions directly was disregarded because it is almost certain that the experimental data themselves are not sufficiently precise to justify any attempt at refinement; moreover, as has been stated previously, any definition of what is to be called *dry air* necessarily involves some exercise of arbitrariness.

The definition of dry air adopted by the International Meteorological Organization differs slightly from that recommended in the preceding paragraph, by including the Paneth[3] values of the concentrations (converted to mol-fractions) of the residual gases. This slight difference will not affect the calculated values of the various thermodynamic properties of dry air, at any rate, not within the accuracy of existing knowledge.

The Working Subcommittee recommends adoption of the International Committee on Atomic Weights (1947) values of the relevant molecular weights[7], namely,

Oxygen (O_2)	32.0000
Nitrogen (N_2)	28.016
Argon (A)	39.944
Carbon Dioxide (CO_2)	44.01
Water (H_2O)	18.016

These yield 28.966, for the apparent molecular weight of dry air.

PHYSICAL CONSTANTS

The Working Subcommittee recommends that each individual zero-pressure reduced free enthalpy $g^\circ_1(T)$ appearing in Equation 1 be computed from best available spectroscopic data on the energy levels of the molecule in question, by the methods of statistical mechanics. Certain fundamental physical constants which have to be determined experimentally and which are therefore subject to experimental error enter these computations. These include: Planck's Constant h, Boltzmann's Constant k, speed of light c, Avogadro's Number N_o, and the relevant molecular weights M_1. For the first four of these the Subcommittee recommends adoption of the Birge[8] (1941) values as follows:

$$h = (6.6242 \pm 0.0024) \times 10^{-27} \text{ erg sec}$$
$$k = (1.380474 \pm 0.00026) \times 10^{-16} \text{ erg/°K}$$
$$c = (2.99776 \pm 0.00004) \times 10^{10} \text{ cm/sec}$$
$$N_o = (6.02283 \pm 0.0011) \times 10^{23} \text{ gmol}^{-1}$$

DuMond and Cohen[9] have treated the various input data from which the values of the above constants have to be computed in a more impartial manner than Birge was able to employ in 1941. Theirs may shortly come to supersede the Birge (1941) values. Fortunately, they leave the combinations kN_o, hcN_o, and hc/k unchanged in value. This means that the only effects of using the DuMond-Cohen (1948) instead of the Birge (1941) values in calculating the thermodynamic properties of moist air would be (a) to reduce the estimated uncertainties of the results and (b) to shift by a constant amount the so-called *absolute* entropy of each constituent. At present, however, the air conditioning engineer, the meteorologist, and the aerodynamicist are interested in *relative* values only of enthalpy and entropy; hence effect (b) is of no practical consequence at present.

CONVERSION FACTORS

The Working Subcommittee also recommends adoption of the following conversion factors:

1.8	°R/°K
30.48	cm/ft
453.5924	g/lb
1	Btu • kg • °K/kcal • lb • °R
860/1.00019	kcal/abs • kwhr
1,013,250	dyne/cm² • atm

The last two of these will bear some explanation. The symbol kcal is intended to designate the so-called *steam tables kilocalorie* defined in the report of the First International Steam Tables Conference[10] as 1/860 (mean) international kilowatt-hour. Birge (1941) recommends the conversion factor 1.00020 ± 0.000045 (mean) international kilowatt-hour per absolute kilowatt-hour. The International Committee of Weights and Measures, at its meetings in October 1946, revised the value of this factor downward to 1.00019. This revision should, of course, be accepted especially since none of the other fundamental constants or conversion factors listed is affected thereby.

According to Birge (1941) the mean density of the mercury in a column 76 cm high at 0 C and 980.665 cm/sec^2 is 13.59504 ± 0.000057 g/cm^3. If the standard atmosphere be defined as the pressure exerted by such a column it would be equal to $1{,}013{,}246 \pm 4$ dyne/cm^2. This is the definition adopted by the International Meteorological Organization. Alternatively, the standard atmosphere is defined in some quarters as equal to 1,013,250 dyne/cm^2, exactly. This is the definition recommended by the Working Subcommittee. The difference between the two atmospheres is exceedingly small but is mentioned here for the sake of definiteness.

ZERO-PRESSURE REDUCED FREE ENTHALPIES

By way of introduction it may be well to mention that, in accordance with relations in Equation 2, zero-pressure enthalpy h^o and zero-pressure reduced entropy s^o (by reduced entropy is meant the sum, $s + R\ln p$) are derivable from zero-pressure reduced free enthalpy g^o (by reduced free enthalpy is meant the difference, $g - RT\ln p$) as follows:

$$h^o = d(\tau g^o)/d\tau \; ; \; s^o = -dg^o/dT$$

Oxygen: Gratch has reduced the recently computed data of Woolley[11] to the following semi-rational equation:

$$g^o/RT = (7/2)\ln\tau + \ln(1 - e^{-\theta\tau}) - A\tau - B\tau^2 - D/\tau - F/(e^{-\theta\tau} - 1) - 1.2164$$

with

$$\theta = 2235.4 \text{ °K}$$
$$A = 1.073 \text{ °K}$$
$$B = 0$$
$$D = 3.30 \times 10^{-6} \text{ °K}^{-1}$$
$$F = 0.0111$$

Values of zero-pressure enthalpy h^o derived from this equation agree with the Woolley data to within a maximum deviation of 0.010 percent and a root mean square deviation of 0.005 percent in the range 100 to 600°K. Values of zero-pressure reduced free enthalpy g^o and zero-pressure reduced entropy s^o show maximum deviations of 0.002 percent and root mean square deviations of 0.001 percent from the Woolley data in this range. These data themselves are probably accurate to within 0.015 percent in the range 100 to 600°K so that a conservative estimate of the probable error of the above equation in this range is 0.02 percent in h^o, g^o, and s^o.

Nitrogen: Gratch[12] has reduced his own calculations of the zero-pressure properties of nitrogen to the following semi-rational equation:

$$g^o/RT = (7/2)\ln\tau + \ln(1 - e^{-\theta\tau}) - A\tau - B\tau^2 - D/\tau - F/(e^{-\theta\tau} - 1) + 0.414686$$

with

$$\theta = 3352.69\ ^\circ K$$
$$A = 0.9580\ ^\circ K$$
$$B = 0.09\ ^\circ K^2$$
$$D = 2.023 \times 10^{-6}\ ^\circ K^{-1}$$
$$F = 0.009009$$

The estimated uncertainty in values of g° calculated from this equation is 0.011 percent in the range 50 to 800°K.

TABLE 1. ZERO-PRESSURE PROPERTIES OF DRY AIR

t °C	h°_a (kcal/kg)	Tol. ±	s°_a (kcal/kg°K)	Tol. ±
-90	43.771	18	1.52160	61
-80	46.165	18	1.53431	61
-70	48.560	19	1.54640	62
-60	50.953	20	1.55791	62
-50	53.348	21	1.56889	63
-40	55.743	22	1.57938	63
-30	58.138	23	1.58944	64
-20	60.533	24	1.59910	64
-10	62.929	25	1.60838	64
0	65.326	26	1.61732	65
10	67.724	27	1.62594	65
20	70.122	28	1.63427	65
30	72.523	29	1.64231	66
40	74.924	30	1.65010	66
50	77.326	31	1.65765	66
60	79.730	32	1.66498	67
70	82.135	33	1.67210	67
80	84.542	34	1.67901	67
90	86.952	35	1.68574	67

TABLE 2. ZERO-PRESSURE PROPERTIES OF WATER VAPOR

t °C	h°_w (kcal/kg)	Tol. ±	s°_w (kcal/kg°K)	Tol. ±
-90	80.290	24	2.28627	69
-80	84.709	25	2.30975	69
-70	89.130	27	2.33207	70
-60	93.552	28	2.35331	71
-50	97.975	29	2.37359	71
-40	102.400	31	2.39299	72
-30	106.828	32	2.41158	72
-20	111.259	33	2.42945	73
-10	115.693	35	2.44661	73
0	120.130	36	2.46317	74
10	124.573	37	2.47914	74
20	129.020	39	2.49457	75
30	133.473	40	2.50951	75
40	137.932	41	2.52399	76
50	142.399	43	2.53803	76
60	146.873	44	2.55166	77
70	151.355	45	2.56492	77
80	155.845	47	2.57780	77
90	160.345	48	2.59037	78

Note: Place rightmost digit of tolerance beneath rightmost digit of value to which it applies, regardless of decimal point location.

Argon: Gratch and Van Voorhis[13] give the following equation valid in the range 50 to 2000°K with a probable error in g° itself of about 0.02 percent:

$$g^\circ/RT = (5/2)\ln\tau - 1.867.$$

Carbon Dioxide: Gratch has reduced the tabulated data of Wagman et al[14] to the following semi-rational equation:

$$g^\circ/RT = (7/2)\ln\tau + 2\ln(1 - e^{-\theta_1/\tau}) + \ln(1 - e^{-\theta_2/\tau}) + \ln(1 - e^{-\theta_3/\tau}) - 1.8945$$

with

$$\theta_1 = 960\ ^\circ K$$
$$\theta_2 = 1944\ ^\circ K$$
$$\theta_3 = 3379\ ^\circ K$$

In the range 300 to 600°K, values of h° and s° derived from this equation agree with the Wagman data within 0.05 percent and 0.005 percent respectively. The probable error of the equation itself is thought to be less than 0.1 percent in this range.

Water Vapor: Goff and Gratch[15] give what amounts to the following semi-rational equation:

$$g^\circ/RT = 4\ln\tau + \ln(1 - e^{-\theta_1/\tau}) + \ln(1 - e^{-\theta_2/\tau}) + \ln(1 - e^{-\theta_3/\tau}) - a_1 e^{-\theta_1/\tau} - a_2 e^{-\theta_2/\tau} - a_3 e^{-\theta_3/\tau} - A\tau - D/\tau + \text{const.}$$

with

$\theta_1 = 2291.16\ ^\circ\text{K}$
$\theta_2 = 5176.37\ ^\circ\text{K}$
$\theta_3 = 5445.59\ ^\circ\text{K}$
$a_1 = -0.03958$
$a_2 = +0.05353$
$a_3 = +0.04000$
$A = 5.011\ ^\circ\text{K}$
$D = 2.32 \times 10^{-5}\ ^\circ\text{K}^{-1}$

Gratch has subsequently adjusted the constant by comparison with the Wagman[14] data as follows:

const. = 4.1083

The probable error of values of g^o calculated from the equation is 0.015 percent in the range −90 to +90°C.

ZERO-PRESSURE PROPERTIES OF DRY AIR

All necessary data are now at hand to compute values of the zero-pressure enthalpy h^o_a and zero-pressure reduced entropy s^o_a for dry air in accordance with the relations,

$$h^o_a = \Sigma x_i h^o_i \text{ and } s^o_a = \Sigma x_i s^o_i \quad \ldots\ldots\ldots\ldots \quad (3)$$

The numerical value of the gas constant R is obtained by multiplying Boltzmann's Constant k into Avogadro's Number N_o and applying appropriate conversion factors from the approved list given earlier in this report; the result is

$$R = 1.98583 \pm 0.00022 \text{ kcal/kmol } ^\circ\text{K}.$$

The apparent molecular weight of dry air is

$$M_a = 28.966 \pm 0.0016 \text{ kg/kmol},$$

the indicated uncertainty arising solely from uncertainties in the molecular weights of its several constituents.

In Table 1 are listed the values so computed, at 10-deg intervals of Centigrade temperature $t(^\circ\text{C})$ defined in terms of absolute temperature $T(^\circ\text{K})$ by the relation,

$$T(^\circ\text{K}) = t(^\circ\text{C}) + 273.16 \quad \ldots\ldots\ldots\ldots \quad (4)$$

Recommended tolerances set at 0.04 percent of the values to which they apply are also listed in Table 1.

ZERO-PRESSURE PROPERTIES OF WATER VAPOR

The necessary data are also at hand to compute the zero-pressure enthalpy h^o_w and zero-pressure reduced entropy s^o_w of water vapor. The present best value of the molecular weight of water is

$$M_w = 18.016 \pm 0.0002 \text{ kg/kmol}.$$

Values of these two properties are listed in Table 2 together with recommended tolerances set at 0.03 percent.

VIRIAL COEFFICIENTS OF DRY AIR

The temperature function $A_{aa}(T)$ is called the *second virial coefficient* of dry air. From Bridgeman's[16] adjustment of the constants in the well-known Beattie-

TABLE 3. SECOND VIRIAL COEFFICIENTS OF DRY AIR

t °C	A_{aa} (cm³/gmol)	Tol. ±	B_{aa} (cm³/gmol)	Tol. ±	C_{aa} (cm³/gmol °K)	Tol. ±
-90	50.4	39	180.6	195	-0.711	107
-80	43.9	33	161.7	167	-0.610	86
-70	38.2	29	145.7	143	-0.529	71
-60	33.2	25	131.9	124	-0.463	58
-50	28.9	22	120.0	108	-0.409	48
-40	25.0	19	109.7	95	-0.363	41
-30	21.6	17	100.6	83	-0.325	34
-20	18.5	15	92.5	74	-0.292	29
-10	15.7	13	85.3	66	-0.265	25
0	13.2	12	78.9	59	-0.240	22
10	10.9	11	73.1	53	-0.220	19
20	8.81	95	67.8	48	-0.201	16
30	6.87	86	63.1	43	-0.1853	142
40	5.09	78	58.7	39	-0.1712	125
50	3.44	71	54.7	36	-0.1586	110
60	1.91	65	51.0	32	-0.1474	98
70	0.49	59	47.6	30	-0.1373	87
80	-0.84	54	44.5	27	-0.1283	77
90	-2.07	54	41.6	25	-0.1201	69

Bridgeman equation of state, its dependence upon absolute temperature T(°K) is given by

$$A_{aa} = -40.70 + 13116\tau + 12 \times 10^{7}\tau^{3} \qquad \text{cm}^3/\text{gmol}.$$

Table 3 lists values of A_{aa} and of the derived coefficients,

$$B_{aa} = d(\tau A_{aa})/d\tau \text{ and}$$
$$C_{aa} = dA_{aa}/dT,$$

together with recommended tolerances obtained by comparison with other formulations.

Strictly speaking the values listed in Table 3 apply to dry, CO_2-free air. The Subcommittee, however, recommends them as applying to dry air as defined in this report because it estimates that the corrections necessary to achieve complete consistency would be well within the probable error of the values themselves.

Third Virial Coefficients

Following the rightmost term in Equation 1 comes a term proportional to the square of the pressure (p^2) in which the so-called *third virial coefficients*

$A_{ijk}(T)$ appear. By analogy, this term is $+\frac{1}{2}p^2\ (2\Sigma x_i x_j x_k A_{ijk} - 3\Sigma_{i,j,k} x_j x_k A_{ijk})$, the value of A_{ijk} being unchanged by permutation of the indices $_{i,j,k}$. The third virial coefficient of dry air $A_{aaa}(T)$ is entirely negligible in the range 150 to 400°K, 0 to 10 atm (atmosphere).

VIRIAL COEFFICIENTS OF WATER VAPOR

Goff and Gratch[17] have formulated the data on the second and third virial coefficients of water vapor as follows:

TABLE 4. SECOND VIRIAL COEFFICIENTS OF WATER VAPOR

t °C	A_{ww} (cm^3/gmol)	Tol. ±	B_{ww} (cm^3/gmol)	Tol. ±	C_{ww} (cm^3/gmol °K)	Tol. ±
-90	-	-	-	-	-	-
-80	-	-	-	-	-	-
-70	-	-	-	-	-	-
-60	-	-	-	-	-	-
-50	-	-	-	-	-	-
-40	-	-	-	-	-	-
-30	-	-	29000	26000	-100	110
-20	-	-	20800	9500	- 71	38
-10	2300	1900	15600	3800	- 51	14
0	1830	800	12000	1600	- 37.2	59
10	1510	400	9440	800	- 28.0	28
20	1260	210	7570	420	- 21.5	14
30	1074	116	6180	230	- 16.84	76
40	924	66	5120	130	- 13.40	42
50	803	40	4298	80	- 10.81	25
60	705	25	3653	49	- 8.85	15
70	625	16	3138	31	- 7.324	90
80	558	10	2722	21	- 6.129	59
90	501	7	2382	14	- 5.179	39

$$A_{ww} = -33.97 + 55306\tau \cdot 10^{72000\tau^2}\ cm^3/gmol$$

$$A_{www} = 0.0348\tau^2 A^3_{ww} cm^3/gmol \cdot atm$$

Values of these and the corresponding derived coefficients, $B = d(\tau A)/d\tau$ and $C = dA/dT$, are listed in Table 4 and Table 5 together with tolerances set at twice the probable error of the values themselves as inferred from the deviations in the experimental data on which they are based. Higher virial coefficients make negligible contributions at pressures below the saturation pressure up to at least 400°K.

INTERACTION COEFFICIENTS OF MOIST AIR

For the purpose of thermodynamic analysis moist air may be regarded as a binary mixture of dry air and water vapor. By this simplification the number of chemical potentials μ_i is reduced to two, namely μ_a and μ_w. Each of these is given as to form by Equation 1; each contains, besides the second virial coefficients A_{aa} and A_{ww}, an interaction coefficient A_{aw}. Goff and Gratch[15] have formulated data obtained under a cooperative investigation between the Towne

Scientific School, University of Pennsylvania, and THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS into the following semi-rational equation:

$$A_{aw} = -29.53 + 0.00669T(1 - e^{-0\tau}) + A\tau + B\tau^2 + D\tau^3 \text{ cm}^3/\text{gmol}$$

TABLE 5. THIRD VIRIAL COEFFICIENTS OF WATER VAPOR

t °C	A_{www} (cm³/gmol atm)	Tol. ±	B_{www} (cm³/gmol atm)	Tol. ±	C_{www} (cm³/gmol atm°K)	Tol. ±
0	-	-	-	-	-	-
10	-	-	-	-	-	-
20	-	-	15000	18000	-47	61
30	-	-	8100	5300	-25	17
40	-	-	4600	1800	-14.0	58
50	-	-	2770	640	- 8.0	20
60	110	120	1710	240	- 4.80	72
70	72	30	1090	100	- 2.97	29
80	48	21	709	43	- 1.87	12
90	33	10	474	20	- 1.21	6

TABLE 6. INTERACTION COEFFICIENTS OF MOIST AIR

t °C	A_{aw} (cm³/gmol)	Tol. ±	B_{aw} (cm³/gmol)	Tol. ±	C_{aw} (cm³/gmol °K)	Tol. ±
-90	84.2	86	226	17	-0.774	93
-80	77.0	82	207	16	-0.674	81
-70	70.7	78	191	15	-0.591	71
-60	65.1	75	177	14	-0.523	63
-50	60.2	72	164	13	-0.466	56
-40	55.8	70	153	13	-0.418	50
-30	51.8	67	143	12	-0.376	45
-20	48.2	65	135	12	-0.341	41
-10	45.0	63	126.6	112	-0.310	37
0	42.0	61	119.5	108	-0.284	34
10	39.3	60	113.0	104	-0.260	31
20	36.8	58	107.0	100	-0.240	29
30	34.5	57	101.6	97	-0.221	27
40	32.3	55	96.5	94	-0.205	25
50	30.4	54	91.9	91	-0.190	23
60	28.5	53	87.6	89	-0.177	21
70	26.8	52	83.6	86	-0.165	20
80	25.2	51	79.9	84	-0.155	19
90	23.7	50	76.4	82	-0.145	17

with

$\theta = 4416.5$ °K
$A = 17{,}546$ cm³ °K/gmol
$B = 95{,}300$ cm³ °K²/gmol
$D = 8.515 \times 10^7$ cm³ °K³/gmol.

Values of A_{aw} and of the corresponding derived coefficients, B_{aw} and C_{aw}, are listed in Table 6 together with tolerances set at thrice the estimated probable errors of the values to which they refer. These probable errors are difficult to estimate reliably because of the meagerness of experimental data; hence they are multiplied by three instead of two to get the corresponding tolerances.

Each chemical potential also contains, besides the third virial coefficients A_{aaa} and A_{www}, the higher order interaction coefficients A_{aaw} and A_{aww}. Pending further research it is necessary to assume that of these four coefficients only A_{www} makes any appreciable contribution to the thermodynamic properties of moist air, at any rate, in the range 150 to 400°K and 0 to 3 atm.

SATURATION PRESSURE OF WATER

Goff and Gratch[17] have calculated the saturation pressure of pure water from other relevant data by means of the identical relations of thermodynamics. Their calculations involve an integration, over the logarithm of absolute temperature, of an integrand which varies only slowly with temperature. Their results may therefore be regarded as determining the function $p_s(T)$ or the function $p_s(t)$, where t is the Centigrade temperature defined by Equation 4.

a. Saturation Pressure of Ice, −100 to 0°C. Values of the saturation pressure of pure ordinary water ice are to be computed from the Goff-Gratch formula,

$$\begin{aligned}\log_{10} p_{si} = {} & -9.09718(T_o/T - 1) \\ & -3.56654 \log_{10}(T_o/T) \\ & +0.876793(1 - T/T_o) \\ & +\log_{10} p_{si}(T_o) \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (5)\end{aligned}$$

with T_o=273.16°K=491.688°R and with $p_{si}(T_o)$=0.0060273 atmospheres, a constant integration which has been adjusted so as to make p_{si} equal to p_{sf} (see following paragraphs) at T=273.17°K, the triple point of water.

The formula as published[17] gives the logarithm of the saturation pressure as the sum of three terms only one of which is explicit in temperature, the other two having been left in the form of definite integrals, over the logarithm of absolute temperature, whose integrands contain both temperature and saturation pressure. In computing numerical values it is easy to evaluate these integrals by the method of successive approximations. For the greater convenience of the user, however, Gratch[18], has replaced these integrals by empirical expressions whose constants he has then adjusted by the method of least squares to make the formula (Equation 5) yield values of $\log_{10}[p_{si}/p_{si}(T_o)]$ showing an average deviation of 0.002 percent and a maximum deviation of 0.015 percent from those given by the formula as previously published.

b. Saturation Pressure of Liquid Water, −50 to 100°C. Values of the saturation pressure of pure ordinary liquid water are to be computed directly from the Goff-Gratch formula[17], namely,

$$\begin{aligned}\log_{10} p_{sf} = {} & -7.90298\ (T_s/T - 1) \\ & +5.02808 \log_{10}\ (T_s/T) \\ & -1.3816 \times 10^{-7}\ (10^{11.344(1-T/T_s)} - 1) \\ & +8.1328 \times 10^{-3}\ (10^{-3.49149(T_s/T-1)} - 1) \\ & +\log_{10} p_{sf}(T_s) \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (6)\end{aligned}$$

with T_s=373.16°K=671.688°R and with $p_{sf}(T_s)$=1 atm.

c. Undercooled Liquid: At temperatures below 0°C it is not possible for the vapor and liquid phases of pure water to coexist in stable equilibrium; it is an experimental fact, however, that they can coexist in what is called *metastable* equilibrium, in which case the coexisting liquid phase is called *undercooled liquid.*

Since reliable experimental data on the thermodynamic properties of undercooled liquid are not presently available, interim values can only be obtained by some sort of extrapolation. Washburn[19] has, in effect, extrapolated the specific heat data of

Barnes and Cooke[20] in the range, −5 to 5°C, to develop a formula for $\log_{10}(p_{wt}/p_{wi})$ which has heretofore been widely used. This formula may now be corrected to make it consistent with the Osborne[21] value of the heat of fusion of ice at 0°C and with the Giauque and Stout[22] measurements of the specific heat of ice.

Gratch[23] has calculated values of the ratio p_{wt}/p_{wi} from: (a) the original Washburn formula; (*b*) the corrected Washburn formula; (*c*) the Osborne, Stimson, and Ginnings[24] formulation of the calorimetric quantity γ using the approximate relation $\gamma = RT^2 d\ln p_{wt}/dT$; (*d*) the corrected Washburn formula adjusted to fit the Scheel and Heuse[25] experimental measurements of p_{wt} down to about −15°C after omitting the last term which is quite small in the experimental range but increases very rapidly as temperature decreases; and (*e*) the Goff-Gratch formulas quoted. The results of these calculations are compared as follows:

SATURATION PRESSURE OF UNDERCOOLED LIQUID

	(p_{wt}/p_{wi})				
t(°C)	(*e*)	(*a*)-(*e*)	(*b*)-(*e*)	(*c*)-(*e*)	(*d*)-(*e*)
0	1.0001	−.0001	0.0000	0.0000	0.0000
−10	1.1023	−.0002	−.0001	+.0005	−.0014
−20	1.2154	−.0004	−.0003	+.0013	−.0018
−30	1.3395	−.0014	−.0009	+.0031	−.0933
−40	1.4738	−.0039	−.0031	+.0072	−.0920
−50	1.6151	−.0077	−.0098	+.0167	+.0044

Another comparison based on an extrapolation of the Smith and Keyes[26] data on the specific volume v_f of the saturated liquid is of interest. Using the Goff-Gratch values of p_{wt} together with previously quoted information regarding zero-pressure enthalpy h^0_w, the virial coefficients A_{ww}, A_{www}, and the corresponding derived coefficients B_{ww}, B_{www}, we may compute values of the specific enthalpy h_f of the saturated liquid from the identical relation, $h_f = h_g - (v_g - v_f)(dp_{wt}/dT)$. It is safe to ignore the extremely small variation with temperature of the difference between h_f and the specific enthalpy h'_w of compressed liquid at atmospheric pressure and regard values of the quantity dh_f/dT as values of the isobaric specific heat c'_w at atmospheric pressure. In the following table these are compared with (*a*) values obtained from extrapolation of the Osborne, Stimson, and Ginnings[24] formula and (*b*) values obtained from extrapolation of the Washburn[19] formulation of the Barnes and Cooke[20] data.

SPECIFIC HEAT OF UNDERCOOLED LIQUID

t(°C)	GOFF-GRATCH	NBS	BARNES AND COOKE
−10	1.018	1.022	1.023
−20	1.039	1.055	1.045
−30	1.075	1.130	1.074
−40	1.14	1.30	1.11
−50	1.27	1.70	1.15

The saturation pressure of undercooled liquid is of such great concern to the meteorologist, if not to the engineer, that it seems necessary to recommend one of these extrapolations pending further research. The International Meteorological Organization

recommends extrapolation of the Goff-Gratch formula down to −50°C because it insures continuity of 0°C and minimizes the number of formulas to be quoted.

Table 7 lists values of saturation pressure calculated from the Goff-Gratch formulas. Recently Keyes[27] has reviewed and reformulated existing data on this important property in the range 0 to 150°C in the light of the findings of Blaisdell and Kaye[28] regarding the relation between the absolute and the International Centigrade temperature scales. His results are in remarkably close agreement with the values listed in Table 7 for temperatures above 0°C.

Regarding the tolerances listed in Table 7, each of those below 0°C has been set at thrice the estimated probable error of the corresponding saturation pressure; those above 0°C are the ones recommended by the Third International Conference on

TABLE 7. SATURATION PRESSURE OF PURE WATER (MILLIBARS[a])

t °C	p_{sf}	Tol. ±	p_{si}	Tol. ±
-90	-	-	0.0000967	9
-80	-	-	0.0005472	44
-70	-	-	0.002615	19
-60	-	-	0.01080	7
-50	0.06355	-	0.03935	22
-40	0.18909	-	0.1283	6
-30	0.50888	-	0.3799	14
-20	1.2543	-	1.032	3
-10	2.863	-	2.597	5
0	6.1078	59	6.107	6
10	12.272	10	-	-
20	23.373	20	-	-
30	42.430	29	-	-
40	73.777	37	-	-
50	123.40	6	-	-
60	199.26	10	-	-
70	311.69	16	-	-
80	473.66	24	-	-
90	701.14	35	-	-

[a] 1 millibar = 1000 dyne/cm²

Steam Tables[29]; no attempt has been made to set tolerances on the recommended values of the saturation pressure of undercooled liquid.

Above 0°C the recommended values of saturation pressure differ slightly, but nevertheless significantly, from those previously recommended by the Third International Conference on Steam Tables[29] if the difference between the absolute and the International Centigrade temperature scales be ignored in making the comparison. The analysis of Keyes[27] may be taken to show, however, that the two sets are brought into remarkably close agreement by transferring the latter from the International Centigrade to the absolute scale by means of the Blaisdell and Kaye[28] findings regarding the relation between the two scales. It seems clear, therefore, that the tolerances recommended by the Third International Conference on Steam Tables could stand reduction.

SATURATION OF MOIST AIR

Moist air is said to be saturated when its condition is such that it can coexist in neutral equilibrium with an associated condensed phase presenting a flat

surface to it. Strictly speaking, the associated condensed phase is not pure water, but water containing small concentrations of dissolved gases. The relative concentrations of these dissolved gases are not the same as those of the same gases in the vapor phase, but we are practically compelled to ignore the very slight differences and to regard the condensed phase as an ideal mixture of pure water and dry air having but two component chemical potentials, μ'_w and μ'_a.

In line with the foregoing simplification we can say that moist air is saturated when it has the same values of temperature, pressure, and its two component chemical potentials as does the associated condensed phase. There are thus four

TABLE 8. THE COEFFICIENT $f_a(p,T)$

	$p = 300$ mb.[a]		$p = 700$ mb.		$p = 1100$ mb.	
t °C	f_{af}	f_{a1}	f_{af}	f_{a1}	f_{af}	f_{a1}
-90	-	1.0027	-	-	-	-
-80	-	1.0024	-	1.0057	-	1.0089
-70	-	1.0022	-	1.0051	-	1.0080
-60	-	1.0020	-	1.0046	-	1.0073
-50	1.0018	1.0018	1.0042	1.0042	1.0065	1.0066
-40	1.0017	1.0017	1.0038	1.0039	1.0060	1.0061
-30	1.0016	1.0016	1.0036	1.0036	1.0055	1.0056
-20	1.0015	1.0015	1.0034	1.0034	1.0052	1.0052
-10	1.0015	1.0015	1.0032	1.0033	1.0049	1.0050
0	1.0016	1.0016	1.0032	1.0032	1.0047	1.0048
10	1.0018	-	1.0032	-	1.0047	-
20	1.0020	-	1.0034	-	1.0048	-
30	1.0023	-	1.0037	-	1.0050	-
40	1.0026	-	1.0041	-	1.0054	-
50	-	-	1.0045	-	1.0059	-
60	-	-	1.0048	-	1.0064	-

[a] 1 millibar = 1000 dyne/cm²

relations to be satisfied at saturation. These four relations involve, however, six independent variables, namely, the two temperatures, the two pressures, and the two mol-fractions; hence, the two-phase system has two degrees of freedom at saturation. This means that such quantities as the weight of water, the volume, the enthalpy, and the entropy—all per unit weight of dry air—must be regarded as functions of two independent variables which are usually chosen to be pressure p and temperature T.

HUMIDITY RATIO AT SATURATION

The weight of water vapor per unit weight of dry air is called *humidity ratio* or *mixing ratio*. It is denoted by the letter W to which may be affixed the subscript $_s$ to specify its value at saturation. It is convenient to introduce a coefficient $f_a(p,T)$ defined by

$$W_s = 0.62197 \frac{f_a p_s / p}{1 - f_a p_s / p} \quad \ldots\ldots\ldots\ldots (7)$$

where p_s is the saturation pressure of pure water, a function of temperature only, and $0.62197 = 18.0160/28.966$.

Goff and Gratch[30, 31] have reduced the conditions for saturation in the case of moist air to

$$\log_e f_s = \alpha(1 - p_s/p) + \beta(p/p_s - 1) \quad \ldots \ldots \ldots \quad (8)$$

with

$$\alpha = (A_{aa} - 2A_{aw} + A_{ww})p_s/RT + A_{www}p_s^2/RT$$
$$\beta = -k'p_s + v'_w p_s/RT + (A_{aa} + 2A_{aw})p_s/RT$$

TABLE 9. COEFFICIENTS IN THE EXPRESSION FOR SATURATION

t °C	α_f	Tol. ±	α_i	Tol. ±	β_f	Tol. ±	β_i	Tol. ±
-90	-	-	0.0000	4	-	-	0.87x10-9	13
-80	-	-	0.0000	4	-	-	0.44x10-8	7
-70	-	-	0.0000	4	-	-	1.90x10-8	29
-60	-	-	0.0000	4	-	-	0.71x10-7	11
-50	0.0000	-	0.0000	4	3.7 x10-7	-	2.35x10-7	35
-40	0.0000	-	0.0000	4	1.02x10-6	-	0.70x10-6	11
-30	0.0001	-	0.0001	4	2.52x10-6	-	1.91x10-6	29
-20	0.0002	-	0.0001	4	0.57x10-5	-	0.48x10-5	7
-10	0.0003	-	0.0003	4	1.21x10-5	-	1.11x10-5	17
0	0.00048	42	0.0005	4	2.37x10-5	36	2.43x10-5	-
10	0.00076	41	-	-	4.44x10-5	67	-	-
20	0.00117	41	-	-	0.79x10-4	12	-	-
30	0.00174	41	-	-	1.34x10-4	20	-	-
40	0.00251	41	-	-	2.19x10-4	33	-	-
50	0.00352	42	-	-	3.46x10-4	52	-	-
60	0.00483	44	-	-	5.26x10-4	79	-	-
70	0.00649	46	-	-	0.78x10-3	12	-	-
80	0.00854	49	-	-	1.12x10-3	17	-	-
90	0.01109	56	-	-	1.58x10-3	24	-	-

where

k' = solubility coefficient of air in water,
v'_w = specific volume of (liquid or solid) water.

Table 8 gives an idea of how the coefficient $f_s(p,T)$ varies with pressure and temperature. In computing the data listed in Table 8, values of k' for liquid water above 0°C have been taken from *International Critical Tables*[32]; k' has been assumed to be zero for undercooled liquid and ice. Values of v'_w for liquid water above 0°C have been taken from Osborne, Stimson, and Ginnings[33]; a constant value of 18.0 cm³/gmol has been assumed for undercooled liquid; values for ice have been taken from Jakob and Erk[34] and from Keenan and Keyes[35].

Rather than list the values of k' and v'_w used in calculating values of the coefficient $f_s(p,T)$ it seems better to list, as is done in Table 9, values of the coefficients α and β in which they appear along with other properties discussed previously. In Table 9 the tolerances given are based on the previously stated tolerances for A_{aa}, A_{aw}, etc., on an arbitrarily chosen tolerance of 10 percent in k', and on zero tolerance in v'_w.

Table 10 lists values of humidity ratio at saturation at standard atmospheric pressure together with tolerances at least twice estimated probable error.

TABLE 10. HUMIDITY RATIO AT SATURATION
(Standard Atmospheric Pressure)

t °C	W_{sf}	Tol. ±	W_{si}	Tol. ±
-90	-	-	0.5991×10^{-7}	66
-80	-	-	0.3386×10^{-6}	34
-70	-	-	0.1617×10^{-5}	14
-60	-	-	0.6677×10^{-5}	53
-50	0.3924×10^{-4}	-	0.2430×10^{-4}	17
-40	1.168×10^{-4}	-	0.7923×10^{-4}	47
-30	0.3141×10^{-3}	-	0.2345×10^{-3}	11
-20	0.7744×10^{-3}	-	0.6370×10^{-3}	26
-10	1.7688×10^{-3}	-	1.6057×10^{-3}	48
0	0.37886×10^{-2}	80	3.7886×10^{-3}	80
10	0.7659×10^{-2}	15	-	-
20	0.14754×10^{-1}	28	-	-
30	0.27319×10^{-1}	49	-	-
40	0.49114×10^{-1}	79	-	-
50	0.8680×10^{-1}	13	-	-
60	0.15340	25	-	-
70	0.27884	47	-	-
80	0.55200	105	-	-
90	1.4160	38	-	-

VOLUME

The correct expression for the specific volume of moist air can be derived from Equation 1 for the chemical potentials by forming the specific free enthalpy g according to the identical relation, $g = \Sigma x_i \mu_i$, and differentiating this according to the identical relation, $v = \delta g / \delta p$. The volume of moist air per unit weight of dry air is obtained from its specific volume in an obvious manner.

TABLE 11. VOLUME OF MOIST AIR PER UNIT WEIGHT OF DRY AIR
(Standard Atmospheric Pressure) (cm^3/g)

t °C	v_a	Tol. ±	v_{sf}	Tol. ±	v_{si}	Tol. ±
-90	517.13	25	-	-	517.13	25
-80	545.69	23	-	-	545.69	23
-70	574.20	22	-	-	574.20	22
-60	602.70	21	-	-	602.71	21
-50	631.18	20	631.22	-	631.20	20
-40	659.65	19	659.77	-	659.73	19
-30	688.09	18	688.44	-	688.35	18
-20	716.53	18	717.42	-	717.26	18
-10	744.96	17	747.06	-	746.88	18
0	773.36	17	778.06	18	778.06	18
10	801.77	17	811.61	19	-	-
20	830.18	17	849.80	21	-	-
30	858.57	18	896.13	25	-	-
40	886.96	18	956.67	29	-	-
50	915.35	19	1042.38	38	-	-
60	943.73	19	1174.90	56	-	-
70	972.10	20	1404.25	93	-	-
80	1000.48	20	1879.2	19	-	-
90	1028.85	21	3341.2	64	-	-

Table 11 lists values of the volume of moist air per unit weight of dry air at standard atmospheric pressure, together with tolerances at least twice the estimated probable error.

ENTHALPY

The correct expression for the specific enthalpy of moist air can be derived from Equation 1 for the chemical potentials by forming the specific free enthalpy g according to the identical relation, $g = \Sigma x_i \mu_i$, and differentiating this accord-

TABLE 12. ENTHALPY OF MOIST AIR PER UNIT WEIGHT OF DRY AIR

(Standard Atmospheric Pressure) (kcal/kg)

t °C	h_a	Tol. ±	h_{af}	Tol. ±	h_{si}	Tol. ±
-90	-21.641	20	-	-	-21.641	20
-80	-19.230	17	-	-	-19.230	17
-70	-16.823	14	-	-	-16.822	14
-60	-14.418	11	-	-	-14.414	11
-50	-12.0132	89	-11.9897	-	-11.9992	90
-40	- 9.6093	68	- 9.5410	-	- 9.5634	71
-30	- 7.2065	49	- 7.0226	-	- 7.0696	55
-20	- 4.8046	31	- 4.3487	-	- 4.4298	46
-10	- 2.4027	16	- 1.3540	-	- 1.4510	45
0	0.0000	0	+ 2.2622	45	+ 2.2622	45
10	2.4021	15	7.0091	105	-	-
20	4.8052	28	13.744	20	-	-
30	7.2096	42	23.880	34	-	-
40	9.6140	55	39.796	54	-	-
50	12.0195	67	65.731	87	-	-
60	14.4264	80	110.00	16	-	-
70	16.8348	91	191.71	31	-	-
80	19.2447	104	367.68	67	-	-
90	21.657	12	921.1	24	-	-

ing to the identical relation, $h = \delta(\tau g)/\delta\tau$, where τ denotes reciprocal absolute temperature. The enthalpy of moist air per unit weight of dry air is obtained from its specific enthalpy in an obvious manner.

The enthalpy of moist air per unit weight of dry air is subject to augmentation by amount $\bar{h}_a + W\bar{h}_w$, where $\bar{h}_a$ and $\bar{h}_w$ are arbitrary constants. In Table 12 the constant $\bar{h}_a$ has been adjusted so as to make the specific enthalpy of dry air zero at 0°C and standard atmospheric pressure; the constant $\bar{h}_w$ has been adjusted so as to make the specific enthalpy of saturated liquid water zero at 0°C in conformity with the usual steam table practice.

ENTROPY

The correct expression for the specific entropy of moist air can be derived from Equation 1 for the chemical potentials by forming the specific free enthalpy g according to the identical relation, $g = \Sigma x_i \mu_i$, and differentiating this accord-

TABLE 13. ENTROPY OF MOIST AIR PER UNIT WEIGHT OF DRY AIR

(Standard Atmospheric Pressure) (kcal/kg°K)

t °C	s_a	Tol. ±	s_{af}	Tol. ±	s_{ai}	Tol. ±
-90	-0.09611	11	-	-	-0.09611	11
-80	-0.08332	9	-	-	-0.08332	9
-70	-0.07116	7	-	-	-0.07116	7
-60	-0.05960	5	-	-	-0.05958	5
-50	-0.04957	4	-0.04846	-	-0.04850	4
-40	-0.03804	3	-0.03774	-	-0.03783	3
-30	-0.02795	2	-0.02716	-	-0.02735	2
-20	-0.01826	1	-0.01638	-	-0.01670	2
-10	-0.00896	1	-0.00478	-	-0.00514	2
0	0.00000	0	+0.00870	2	+0.00870	2
10	+0.00864	1	0.02574	4	-	-
20	0.01698	1	0.04911	7	-	-
30	0.02504	2	0.08312	12	-	-
40	0.03284	2	0.13485	18	-	-
50	0.04040	2	0.21669	29	-	-
60	0.04774	3	0.35246	52	-	-
70	0.05486	3	0.59601	95	-	-
80	0.06178	4	1.11236	203	-	-
90	0.06852	4	2.67830	71	-	-

ing to the identical relation, $s = -\delta g/\delta T$. The entropy of moist air per unit weight of dry air is obtained from its specific entropy in an obvious manner.

The entropy of moist air per unit weight of dry air is subject to augmentation by amount $\bar{s}_a + W\bar{s}_w$, where $\bar{s}_a$ and $\bar{s}_w$ are arbitrary constants. In Table 13 the constant $\bar{s}_a$ has been adjusted so as to make the specific entropy of dry air zero at 0°C and atmospheric pressure; the constant $\bar{s}_w$ has been adjusted so as to make the specific entropy of saturated liquid water zero at 0°C in conformity with the usual steam table practice.

TABLE 14. PROPERTIES OF CONDENSED WATER

(Standard Atmospheric Pressure)

t °C	h_w liquid	h_w solid	p_w liquid	p_w solid
-90	-	-117.79	-	-0.4590
-80	-	-114.25	-	-0.4402
-70	-	-110.54	-	-0.4214
-60	-	-106.64	-	-0.4028
-50	-53.97	-102.58	-0.2192	-0.3841
-40	-42.00	- 98.34	-0.1666	-0.3655
-30	-30.96	- 93.92	-0.1201	-0.3470
-20	-20.40	- 89.34	-0.0776	-0.3285
-10	-10.14	- 84.57	-0.0378	-0.3101
0	+ 0.02	- 79.64	0.0000	-0.2916
10	10.06	-	+0.0361	-
20	20.06	-	0.0708	-
30	30.04	-	0.1043	-
40	40.03	-	0.1367	-
50	50.01	-	0.1681	-
60	60.00	-	0.1985	-
70	69.99	-	0.2281	-
80	80.01	-	0.2568	-
90	90.05	-	0.2848	-

CONDENSED WATER

In the definition of thermodynamic wet bulb temperature to be given later there appears the specific enthalpy h'_w of pure (air-free) compressed liquid (or solid) water. Values of h'_w and of the corresponding specific entropy s'_w are listed in Table 14 for standard atmospheric pressure. The reference point at which both enthalpy and entropy are assigned the value zero is saturated liquid at 0°C in conformity with usual steam table practice. The values for undercooled liquid are based in part on an extrapolation of the Smith and Keyes[26] volume data, in part on the recommended extrapolation of the Goff-Gratch formula in Equation 6 for the saturation pressure p_{wt}, and in part on other data previously quoted. Since the first extrapolation is a fairly safe one inasmuch as the final results depend only slightly on the liquid volume, these values should be recommended for adoption, pending further research, if for no other reason than that they are consistent with previous recommendations.

REFERENCE POINTS

The Working Subcommittee discussed the question of reference points at some length. Moist air, as defined in this report, is a mixture of five constituent gases: oxygen (O_2), nitrogen (N_2), argon (A), carbon dioxide (CO_2), and water vapor (H_2O). The enthalpy of each of these constituent gases i is subject to augmentation by a constant amount $\bar{h}_i$, called its null-point ($p = 0$, $T = 0$) enthalpy; the entropy of each constituent is subject to augmentation by a constant amount $\bar{s}_i$, called its null-point reduced entropy (by reduced entropy is meant the sum, $s + R\ln p$, with pressure expressed in standard atmospheres). It follows that the enthalpy and entropy of dry air are subject to augmentation by constant amounts, $\bar{h}_a = \Sigma x_i \bar{h}_i$ and $\bar{s}_a = \Sigma x_i \bar{s}_i$, respectively; similarly, the enthalpy and entropy of water are subject to augmentation by constant amounts, $\bar{h}_w$ and $\bar{s}_w$, respectively.

Tables 1 and 2 of this report are based on the arbitrary assignments: $\bar{h}_a = \bar{s}_a = \bar{h}_w = \bar{s}_w = 0$.

Tables 12, 13, 14 are based on the following assignments: for dry air, both enthalpy and entropy are assigned the value zero at 0°C, atmospheric pressure; for water, both enthalpy and entropy are assigned the value zero at 0°C, saturated liquid. Using the data previously quoted in this report it is easy to compute that these assignments are to be implemented by making: $\bar{h}_a = -65.260$ kcal/kg, $\bar{s}_a = -1.61712$ kcal/kg°K, $\bar{h}_w = +477.277$ kcal/kg, $\bar{s}_w = -0.83960$ kcal/kg°K.

The Goff-Gratch tables, Thermodynamic Properties of Moist Air[15] and Thermodynamic Properties of Water at Saturation[17], are based on the following assignments: for dry air, both enthalpy and entropy are assigned the value zero at 0°F, atmospheric pressure; for water, both enthalpy and entropy are assigned the value zero at 32 F, saturated liquid. Using the data previously quoted in this report it is easy to compute that these assignments are to be implemented by making: $\bar{h}_a = -109.782$ Btu/lb, $\bar{s}_a = -1.60096$ Btu/lb°R, $\bar{h}_w = +859.099$ Btu/lb, $\bar{s}_w = -0.83960$ Btu/lb°R. The tables referred to have

THERMODYNAMIC PROPERTIES OF MOIST AIR

STANDARD ATMOSPHERIC PRESSURE

t(°C)	$\bar{h}_a$	$\bar{h}_s$	$\bar{s}_a$	$\bar{s}_s$	$\bar{h}'_w$	$\bar{s}'_w$
−90	43.619	43.619	1.52101	1.52101	−595.07	0.3806
0	65.260	65.714	1.61712	1.62030	−477.26	0.8396
90	86.917	310.54	1.68564	2.87451	−387.23	1.1244

been reproduced in HEATING, VENTILATING, AIR CONDITIONING GUIDE published and widely distributed by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Using the same assignments, additional tables giving the thermodynamic properties of moist air for six different pressures above atmospheric and up to 150 lb/in.^{2} have been compiled in the University of Pennsylvania Thermodynamics Research Laboratory[36] under Project G-8. It is clear, therefore, that the assignments of this paragraph are destined to be used rather widely in engineering circles.

On the other hand, the tables being compiled in the University of Pennsylvania Thermodynamics Research Laboratory for publication in Smithsonian Meteorological Tables[31] are based on the following assignments: for dry air, both enthalpy and reduced entropy (by reduced entropy is *here* meant the sum, $s + R\ln p$, with pressure expressed in *millibars*) are assigned the value zero at 0°C, zero pressure; for water, both enthalpy and entropy are assigned the value zero at 0°C, saturated liquid. Using the data previously quoted in this report it is easy to compute that these assignments are to be implemented by making: $\bar{h}_a = -65.326$ kcal/kg, $\bar{s}_a = -2.09179$ kcal/kg°K, $\bar{h}_w = +477.277$ kcal/kg, $\bar{s}_w = -0.83960$ kcal/kg°K.

In most, if not all, tables giving the thermodynamic properties of moist air, the usual steam tables practice of assigning the value zero to both the enthalpy and the entropy of water at 0°C, saturated liquid is followed. There is, however, no unanimity in the choice of reference point for dry air. Perhaps this lack of unanimity is of little practical importance because it is so easy to convert from one choice to another. The Working Subcommittee decided not to indicate a preference.

Recent developments in the science of interpreting atomic and molecular spectra have focused attention upon the zero-pressure isobar as the line along which the thermodynamic properties of gases and gas mixtures generally can at any time be known with greatest accuracy. This is certainly true in the case of moist air; hence the relative permanence of the numerical values assigned to the null-point enthalpies and reduced entropies of dry air and water would be enhanced by placing both reference points on the zero-pressure isobar at some stated value(s) of absolute temperature. The obvious location would be the null-point ($p = 0$, $T = 0$) itself; indeed the null-point suggests itself as the logical reference point for all gases.

Even if the null-point were to be adopted as the common reference point it would still be necessary to decide what are the best values to assign to the enthalpy and reduced entropy there for each gas under consideration. Tables 1

and 2 of this report are based on the assignments, $\bar{h}_a = \bar{s}_a = \bar{h}_w = \bar{s}_w = 0$, as is the preceding skeleton table.

In this table the unit of enthalpy is the kcal/kg, that of entropy, the kcal/kg°K. At −90°C the values given for h_s and s_s are for saturation with respect to ice; those for h'_w and s'_w at this temperature are for ice also. The data in this table should be compared with those in Tables 12, 13, and 14.

According to the so-called Third Law of Thermodynamics the assignments, $\bar{s}_i = 0$, are always permissible if nuclear changes are ruled out of consideration provided that, in computing values of $s^o_i - \bar{s}_i$, possible contributions from nuclear spin and other nuclear energy levels are consistently included or excluded (it is common practice to exclude). But the assignments, $\bar{h}_i = 0$, are in general not permitted if ordinary chemical changes are not to be ruled out of consideration. In this case only as many of the $\bar{h}_i$ can be disposed of arbitrarily as there are distinct atomic species. It would therefore seem logical to exercise this permitted arbitrariness on the null-point enthalpies of the monatomic gases themselves. Undoubtedly there will be a move in this direction as the accuracy and completeness of our knowledge of the dissociation energies of molecular gases, by which their null-point enthalpies are related to those of the associated monatomic gases, become more satisfactory.

DEW-POINT TEMPERATURE

A derived thermodynamic property of moist air of great importance in hygrometry is the so-called thermodynamic dew-point temperature T_d. The Working Subcommittee recommends that this important property be defined as the solution $T_d(p,W)$ of the equation,

$$W_s(p,T_d) = W \quad \ldots\ldots\ldots\ldots\ldots \quad (9)$$

The definition is given in mathematical form to avoid ambiguity. It is a problem belonging properly to practical hygrometry to design and construct dew point hygrometers capable of reading values of thermodynamic dew-point temperature as defined by Equation 9 after suitable calibration. The I.M.O. has concurred in the foregoing definition.

WET BULB TEMPERATURE

Another derived thermodynamic property of moist air of great importance in psychrometry is the so-called thermodynamic wet bulb temperature T_w. The Working Subcommittee recommends that this important property be defined as the solution $T_w(p,T,W)$ of

$$h(p,T,W) - W \bullet h'_w(p,T_w) = h_s(p,T_w) - W_s(p,T_w) \bullet h'_w(p,T_w) \quad \ldots\ldots \quad (10)$$

where it is to be understood that h'_w denotes the specific enthalpy of pure compressed liquid (or solid) water. This definition is also given in mathematical form to avoid ambiguity. It is a problem belonging properly to practical psychrometry to design and construct psychrometers capable of reading values of calibration. It seems likely that the I.M.O. will concur in the foregoing definition.

In justification of this definition it may be said that it combines the correct energy and weight accountings for the steady-flow process of injecting pure compressed liquid (or solid) water at pressure p and temperature T_w into a stream of moist air at pressure p, temperature T, and humidity ratio W to bring the air adiabatically to saturation at pressure p and temperature T_w. This is to be regarded as the appropriate idealization of the actual process by which the thin film of water on a wet bulb thermometer immersed in a stream of moist air maintains, at any rate for a time, a steady value below that of the air itself.

ACKNOWLEDGMENT

It is a pleasure to acknowledge the substantial cooperation of the Navy Department, Bureau of Ships, in the preparation of this report. It was under the sponsorship of the Bureau that the University of Pennsylvania Thermodynamics Research Laboratory was operated from July 1, 1945, until June 30, 1948, when cognizance was transferred to the Office of Naval Research. The invaluable aid of S. Gratch, assistant professor of Mechanical Engineering, Towne Scientific School, University of Pennsylvania, and Project Leader in charge of gas properties research in the University of Pennsylvania Thermodynamics Research Laboratory is also gratefully acknowledged.

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No. 1376

SPECIAL MEETING

NEW YORK, N. Y.—NOVEMBER 21, 1949

Pres. A. E. Stacey, Jr., called a special meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to order at 6:00 p.m., November 21, 1949, at the Building Trades Club, 2 Park Ave., New York, N. Y.

The Secretary, A. V. Hutchinson, read the call for the meeting:

1. To act upon a proposal to revise the Charter (Certificate of Incorporation) of the Society by substituting for the present purposes and powers the attached purposes, powers, and provisions, and to authorize the Officers of the Society to execute and file with the proper authorities an appropriate Certificate of Amendment.

2. To consider the written proposal to amend the Constitution, By-Laws and Rules of the Society by substituting therefor a new set of By-Laws as recommended by the Constitution and By-Laws Committee and approved by the Council; and to determine whether the same shall be mailed to the members of the Society for action at the Society's next Annual Meeting.

President Stacey said that each member had received the meeting notice and copy of the proposed Charter revisions by mail and he would entertain a motion authorizing the Officers of the Society to take the necessary action to file amendments to paragraph 4 of our Charter (Articles of Incorporation) in order to improve our situation as a scientific and educational organization.

M. F. Blankin, Philadelphia, Pa., offered the following resolution which was seconded by Ernest Szekely, Milwaukee, Wis.:

RESOLVED, THAT in place of Paragraph 4 on the Certificate of Amendment filed in the office of the Secretary of State on or about March 8, 1946, the following be substituted:

(*a*) To advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

(*b*) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, the results of which shall be made freely available to the public.

(*c*) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, reports,

educational programs and scientific and technical data relating to heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(*d*) To engage in educational activities (not including the conduct of any school, schools or institutions of learning), and to encourage the adoption and maintenance of high standards of instruction and technical and professional training in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(*e*) To cooperate with governmental agencies and with universities, colleges, schools and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants and awards in furtherance of the foregoing purposes.

(*f*) To assist in the formation of local chapters and student and other branches, and to regulate, operate and control the same under the direction and at the pleasure of the corporation, but no local chapter or branch shall subject the corporation to any financial or other obligation except such as the corporation may voluntarily assume.

(*g*) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount or value, for any of the corporation's objects, by way of bequest, devise, gift, purchase or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest or principal.

(*h*) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(*i*) This corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of this corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation. In the event of the dissolution of the corporation, the Council (directors) shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer or director of the corporation.

As there was no discussion of the motion, the vote was taken, and the motion was adopted unanimously. (In addition to the members present, 13 proxies were voted in favor of the amendments.)

President Stacey then said that the second item of business would be considered, namely, the submission of the proposed By-Laws to the membership with any changes made at this meeting to be submitted for ratification at the 56th Annual Meeting in January, 1950.

President Stacey said that about 2 years ago, Council had requested the Constitution and By-Laws Committee to revise and streamline our Constitution, By-Laws, Rules and Regulations, and bring them in line with present practice and operation of the Society giving due consideration to resolutions passed by Council, reports of Officers and various Committee recommendations. Legal counsel was retained to review the work with the Committee so as to meet all legal requirements of our Charter.

When the Committee completed its work, its report was sent to Council for discussion at its meeting in Baltimore, October 16 and 17, 1949, accompanied by the comments of the Committee.

The Council spent nearly 2 days analyzing the report, article by article, and it recommended some changes.

On motion of Mr. Blankin seconded by Mr. Szekely,

IT WAS VOTED to have the Secretary read the memorandum of the Constitution and By-Laws Committee, which summarized the changes contained in the report.

This was done by the Secretary.

C. F. Boester, Lafayette, Ind., stated that he wanted to make it a matter of record as to whether or not they had the opportunity to review it and whether or not they had the privilege of voting and determining whether or not they are in complete agreement as to the changes made by the Council at the Baltimore meeting.

In reply to a question by R. H. Carpenter, New York, N. Y., as to why the word, *Constitution,* was omitted from the new document, it was explained that under the Membership Corporation law of the State of New York, all Rules, Regulations, Constitutions are regarded as By-Laws.

L. N. Hunter, Johnstown, Pa., pointed out that it was his feeling that it was not advisable to change the present method of handling research payments and he thought the phrase, *or authored,* in **Article VII,** *Section 3 (h)*, should be eliminated. He questioned the need for haste in adopting these proposed By-Laws.

In answer, it was pointed out that the Certificate of Incorporation was the essential part, but the adoption of the new By-Laws would eliminate a tremendous amount of detail and duplication which makes for confusion. The only substantial change was in the centralizing of finances. **Article VII,** *Section 3 (h)*, paragraph 6, concerning disbursement of research funds, was the result of conferences among the Council, the Director of Research, the accountant and the Secretary. It was deemed to be the most practical solution.

Cyril Tasker felt that there was room for discussion as to interpretation of some of the Rules promulgated in the new By-Laws. He wanted more time for study and added that he had changed his mind concerning the adoption of **Article VII,** *Section 5 (h)* concerning the disbursement of funds for research.

Mr. Flink questioned the all-inclusiveness of the section dealing with formulae, discoveries, inventions, etc., of employees of the Society being property of the Society. He felt the sentence as written even included inventions outside of Society interests. He desired the insertion of words, *matter which shall be of interest to the Society.*

L. E. Seeley pointed out that the courts would never permit the Society to overreach itself. Legal counsel stated that the inclusion of limiting phrases on inventions were not necessary because the Charter and By-Laws of which this sentence is a part indicate the particular field in which the Society is engaged. Mr. Tasker asked whether this clause covered employees not directly engaged in research. The answer was, *This clause affects only research employment.*

On motion of Mr. Hunter, seconded by Mr. Blankin,

IT WAS VOTED THAT the words *or authored* be omitted from **Article VII,** *Section 5 (h)*, paragraph 6.

In response to a member's question, President Stacey said that this meeting was not to pass these By-Laws but only to vote on their submission to the Annual Meeting, and all members would receive copies of the proposed By-Laws as amended here to study in advance of the Meeting. President Stacey reminded the members that failure to submit these prepared By-Laws to the membership would mean another delay with the instituting of the procedure all over again. By adopting these new By-Laws, the Society would legally be classified as a pure scientific and educational society. Moneys donated or bequeathed the Society would be non-taxable gifts.

Mr. Blankin reminded the members that the Constitution and By-Laws Committee had studied the proposed document for over two years, and the membership would probably be able to study the proposed document in a period of six weeks. Little would be accomplished in a six month period, he went on, and the Society would be a year further behind in the adoption of the By-Laws.

Mr. Boester urged immediate action. He pointed out that although there were points of disagreement on the wording of the By-Laws, that there had been sufficient study by those competent to study the document, and he proposed voting for it, and overlooking some of the minor objections.

A. A. Bearman said that there was provision for Amendment if the necessity for it arose and Mr. Blankin added that this meeting was not adopting the By-Laws, but only debating the submission of them to the Annual Meeting. It was the opinion of President Stacey that, a great deal of the work of our organization has to be done by Committees, because that is the only way we can carry on Society business.

Mr. Blankin's view was that a rejection now would mean another year's delay and he expressed doubt that many members would study the By-Laws. Mr. Blankin called for the question and

IT WAS VOTED THAT the proposed By-Laws with changes approved at this Special Meeting be submitted to the members for action at the Annual Meeting in Dallas, January 23-26, 1950.

R. H. Carpenter suggested that the Secretary when mailing the meeting notice and a copy of the proposed By-Laws to the members, enclose an explanatory letter indicating the reason for the short notice. He desired that the membership be made to realize the importance of not having another year pass before adoption of these By-Laws.

President Stacey approved the suggestion and Mr. Boester as chairman of the Chapter Relations Committee promised to bring these proposed By-Laws to the attention of the Chapter officers and urge them to bring them up in their Chapter meetings for discussion before the Annual Meeting.

On motion of Mr. Boester, seconded by Mr. Carpenter,

IT WAS VOTED THAT the meeting be adjourned at 7:15 p.m.

CHARTER

Provisions of
CERTIFICATE OF INCORPORATION
of
THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

(As filed on January 24, 1895 and amended by Certificates of Amendment filed on or about May 20, 1914, March 8, 1946 and December 7, 1949)

1. The corporate name of the Society is to be known as THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

2. The certificate of incorporation was filed in the office of the Secretary of State on or about January 24, 1895. Amended certificates of incorporation were filed in the office of the Secretary of State on or about May 20, 1914, March 8, 1946 and December 7, 1949.

3. The number of directors shall be not less than thirteen (13) nor more than twenty-one (21). The Board of Directors shall be styled "The Council."

4. The purposes for which this corporation is organized are:

(*a*) To advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

(*b*) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, the results of which shall be made freely available to the public.

(*c*) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, reports, educational programs and scientific and technical data relating to heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(*d*) To engage in educational activities (not including the conduct of any school, schools or institutions of learning), and to encourage the adoption and maintenance of high standards of instruction and technical and professional training in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

Adopted at a Special Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held in New York, N. Y., November 21, 1949.

(*e*) To cooperate with governmental agencies and with universities, colleges, schools and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants and awards in furtherance of the foregoing purposes.

(*f*) To assist in the formation of local chapters and student and other branches, and to regulate, operate and control the same under the direction and at the pleasure of the corporation, but no local chapter or branch shall subject the corporation to any financial or other obligation except such as the corporation may voluntarily assume.

(*g*) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount or value, for any of the corporation's objects, by way of bequest, devise, gift, purchase or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest or principal.

(*h*) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(*i*) This corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of this corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation; in the event of the dissolution of the corporation, the Council (directors) shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer or director of the corporation.

5. The principal place of business or office of said Society is to be in the City and County of New York.

(*a*) The territory in which the corporation's operations are principally to be conducted is in all parts of the United States and its possessions and in the Dominion of Canada, but the corporation may on occasions extend its operations to other parts of the world.

(*b*) No officer, director or member of this corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in effecting one or more of its purposes as the Board of Directors may determine.

(*c*) Each director of the corporation shall be indemnified by the corporation against expenses actually and necessarily incurred by him in connection with the defense of any action, suit or proceeding in which he is made a party by reason of his being or having been a director of the corporation, except in relation to matters as to which he shall be adjudged in such action, suit or proceeding to be liable for willful negligence, misfeasance or misconduct in the performance of his duties as director; such right of indemnification shall not be deemed exclusive of any other right to which he may be entitled under any by-law, agreement, vote or otherwise.

In Memoriam 1949

Name	Joined
Joseph C. Albright, Baltimore, Md.	1946
George W. Barr *(Life Member),* Philadelphia, Pa.	1905
John Bauer, Plattsmouth, Nebr.	1946
Henry A. Baxter, Toronto, Ont., Canada	1944
J. Herbert Bekeshus, Troy, N. Y.	1945
Louis H. Bergeron, Toronto, Ont., Canada	1949
Frederick R. Bishop, Detroit, Mich.	1921
Edward J. Calnan, St. Catherines, Ont., Canada	1941
Albert A. Criqui *(Life Member),* Buffalo, N. Y.	1919
Alfred J. Edge, San Mateo, Calif.	1938
Meryl V. Eicholtz, Milwaukee, Wis.	1942
Gosta E. Ekholm, Stockholm, Sweden	1949
Harry W. Ellis *(Life Member),* Milwaukee, Wis.	1909
W. R. Etie, Houston, Tex.	1943
Edward J. Farrell, Detroit, Mich.	1943
Wilbur R. Fitch, Belmont, Mass.	1945
John J. Foley, Toronto, Ont., Canada	1938
Michael H. Gornston *(Life Member),* Forest Hills, N. Y.	1923
William Z. Harrison, Salt Lake City, Utah	1944
A. J. Harton, St. Joseph, Mo.	1935
Harland B. Hinckley, Chicago, Ill.	1934
William T. Hopson *(Life Member),* New London, Conn.	1915
John Howatt *(Life Member* and *Presidential Member),* Chicago, Ill.	1915
Vernon E. Hupp, Chicago, Ill.	1949
John A. Larkin, New Haven, Conn.	1947
John J. Lawlor, Toronto, Ont., Canada	1935
Thomas E. McGrail, Ottawa, Ont., Canada	1926
Carle J. Merrill *(Life Member),* Portland, Me.	1919
Walter A. Mertz, Chicago, Ill.	1919
James M. Muckle, Hong Kong, China	1939
Louis S. Murphy, Jr., Chevy Chase, Md.	1949
Andrew G. O'Rear, Los Angeles, Calif.	1930
Edwin J. Ostermeier, Chicago, Ill.	1945
Rudolf Pollack, New York, N. Y.	1937
B. F. Raber, Berkeley, Calif.	1937
Noah M. Simmons, Indianapolis, Ind.	1944
Howard R. Stevens, Reno, Nev.	1941
Sherman W. Strouse, Buffalo, N. Y.	1949
James Terry, Detroit, Mich.	1948
Herbert G. Thomas, Chicago, Ill.	1917
George J. Wallace, Elmhurst, N. Y.	1923
W. A. Weber, Minneapolis, Minn.	1947
J. Walter Williams* *(Life Member),* Ithaca, N. Y.	1915

* Died in 1948 but Society notified in 1949.

JOHN HOWATT, Chicago, Ill.

John Howatt

October 22, 1880—October 22, 1949

John Howatt, who became a Life Member of the Society in 1945 and served as president of the Society in 1935, died on October 22, after an illness of several months. His passing is mourned by his many friends in the Society and by his associates in the City of Chicago, where he served the past 36 years as chief engineer of the Board of Education and also as its business manager for the last five years.

The Officers and Council feel keenly the loss of a loyal and active member who was ever ready to serve, a splendid engineer, and a sincere friend.

Mr. Howatt was born in Banff, Scotland, in 1880, and received his university education in the United States, graduating from the University of Minnesota in 1904. His career in engineering began with shop experience at the Westinghouse Electric and Manufacturing Co., East Pittsburgh, and later he joined the U. S. Navy Department in Seattle, Wash. Subsequently he became an engineering inspector of mechanical and electrical equipment for the U. S. Treasury Department. For a brief time he was a consulting and contracting engineer in Seattle.

From 1913 until his death he was chief engineer of the Board of Education and supervised the design, installation, and operation of mechanical and electrical equipment in Chicago public schools.

A member of the Society since 1915, Mr. Howatt had been very active in its affairs, both local and national. He served on many Society committees such as: the Committee on Schoolhouse Standards; Committee on Constitution and By-Laws; Committee on Engineering and Its Relation to Public Health; Committee on Relation of Body Changes to Air Changes; F. Paul Anderson Award Committee; A.S.H.V.E.-*A.S.R.E.* Committee on National Standards for Air Conditioning Applications; Chairman of Exposition Advisory Committee; Guide Publication Committee; Chairman of TAC on Air Flow through Registers and Grilles; Committee on Chapter Relationship; Committee on Standards of Ventilation; Committee to Cooperate with Rochester School Board. He became a member of Council in 1927 and served on its Executive, Membership, Finance, and Meetings Committees, and on the Advisory Board. He also served as First Vice President and Second Vice President. In 1920 he was President of the Illinois Chapter and was an active worker until his death.

Mr. Howatt wrote extensively on engineering and philosophical subjects and was on the board of consulting and contributing editors of *Heating, Piping & Air Conditioning* for the past 20 years.

He was affectionately known to his associates as "Chief," was always ready and willing to be helpful to his fellowman and will long be remembered for his great interest in the advancement of his profession, the unselfish service he rendered to the Society and in civic affairs.

A fitting milestone in his service to the Society was the occasion of the 55th Annual Meeting in Chicago in January 1949 when he served as an honorary chairman of the Committee on Arrangements and as toastmaster at the banquet. His legion of friends in the Society and in other fields of activity will miss his genial presence and his keen sense of humor.

Mr. Howatt is survived by his wife, Della, his son, Gordon, his grandchildren, John and Patricia, his brother, William, and his sister, Mrs. Isabel Coulter. Interment was at Lake City, Minn., on Wednesday, October 26.

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